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Helsinki University of Technology. Department of Mechanical Engineering.
Laboratory of Heating, Ventilating and Air Conditioning. A
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THE EFFECT OF THERMAL PLUMES ON THE PERFORMANCE OF VENTILATED CEILINGS IN COMMERCIAL KITCHENS

REPORT A11

Risto Kosonen



TEKNILLINEN KORKEAKOULU
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HELSINKI UNIVERSITY OF TECHNOLOGY

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ABSTRACT

The efficiency of the exhaust system is especially emphasised with a ventilated ceiling system where the exhaust is located at ceiling level. The removal efficiency of the total system must be guaranteed and the spread of impurities throughout the kitchen should be prevented. At the moment, none of the existing calculation standards are specially tailored for a kitchen ceiling environment. In the normal design practice, empirical knowledge of the existing installations together with heat load based calculation has been used for airflow rate determination.

The starting point for this research was to study the effect of the thermal plumes and supply air systems on the efficiency of a ventilated ceiling. A special consideration was to analyze the effect of a capture jet on the contaminant removal efficiency. In that capture air concept, the air jet is projected horizontally across the ceiling, which helps to direct heat and air impurities towards the exhaust. From the practical point of view, the objective of this study was to develop a design process to compute the required air flow rate more accurately.

In this study, the measured convection flows of kitchen appliances during idle and cooking modes were compared with the generic plume equation in which the virtual origin is constant. The generic plume equation derived in the region of complete flow similarity is not fully valid in the intermediate zone (0.8-2.0 m from appliances). Still, it is possible to reach a reasonable accuracy for practical applications with the adjusted virtual origin. The cooking process does not have any significant effect on the velocity and temperature distribution of the convection flow. The reason for this is that the mass flow rate of water during boiling is small compared with the induced air flow rate and therefore does not have a significant effect on the convection flow. Thus, the actual convection load and the product specific virtual origin can describe the plume during the cooking process.

In the previous studies of thermal plumes, the velocity and temperature distribution factors are much higher than in the present study. In addition, the entrainment factors of the previous studies are much smaller than in this study. The measurements indicate that the heat gain has a significant effect on the spreading angle and the entrainment factor close to the heat source. The plumes with high heat gains are narrower and the convection flow induces more room air than the previous studies have pointed out.

The efficiency of the exhaust system can be improved with a small capture jet installed at the ceiling surface. Both the measurement and simulated data give lower contaminant levels when the capture jet was introduced. The plume equation gives a platform to calculate the air flow rate that is theoretically required to remove the convective heat output of the appliance block. In this study, the flush-out factor of the supply air on the theoretical plume equation was derived for the centralized capture jet concept. For practical design work, the target for the containment removal efficiency should be 85 %. To obtain 85 % containment removal efficiency requires to a flush-out factor of 1.2

PREWORD

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I wish to express my gratitude to Panu Mustakallio who conducted CFD- simulations for this work.

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Risto Kosonen
Vantaa in May 2006

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LIST OF PUBLICATIONS

- I. Kosonen Risto, Koskela Hannu, Saarinen Pekka; Thermal plumes of kitchen appliances: part 1 idle mode (accepted on 29.12.2005 Energy and Buildings)
- II. Kosonen Risto, Koskela Hannu, Saarinen Pekka; Thermal plumes of kitchen appliances: part 2 cooking mode (accepted on 29.12.2005 Energy and Buildings)
- III. Kosonen Risto, Koskela Hannu, Saarinen Pekka; An analysis of thermal plumes. Proceedings of 17th of air-conditioning and ventilation conference May 17-19 2006 in Prague Czech Republic.
- IV. Kosonen Risto, Mustakallio Panu; The influence of a capture jet on the efficiency of a ventilated ceiling in a commercial kitchen. The International Journal of Ventilation. Volume 1 Number 3 Feb.2003 pages 189-200
- V. Kosonen Risto, Mustakallio Panu; Analysis of Capture and Containment Efficiency of a Ventilated Ceiling. The International Journal of Ventilation. Volume 2 Number 1 Jun.2003 pages 33-44
- VI. Kosonen Risto The effect of supply air systems on the efficiency of a ventilated ceiling. (accepted on 5.9.2005 Building and Environment.)

The author is the principle author of six publications. All authors made the analysis and the conclusions jointly.

NOMENCLATURE

| | |
|-------|--|
| a | product specific factor |
| Ar | Archimedes' number |
| c_p | specific heat at constant pressure (kJ/kg,K) |
| c | pollution concentration (kg/m ³) |
| D_h | hydraulic diameter (m) |
| E | kinetic energy flux (W) |
| g | gravitational acceleration (m/s ²) |
| H | enthalpy flux (W) |
| K | factor |
| k | installation factor of the kitchen appliance in plume calculation, turbulent kinetic energy |
| L | length of a heat source (m) |
| m | velocity distribution factor |
| M | momentum flux (kgm/s ²) |
| P | point |
| p | temperature distribution factor |
| q_v | air flow rate (m ³ /s) |
| r | radial distance (m) |
| R | profile width where 1/e of the maximum is obtained (m) |
| S | emission (kg/s) |
| T | temperature (°C) |
| v | velocity (m/s) |
| W | width of a heat source (m) |
| z | vertical distance (m) |

Greek symbols

| | |
|---------------|---|
| α | entrainment factor |
| β | thermal expansion coefficient (1/K) |
| Δ | difference |
| ε | turbulence dissipation rate |
| Φ | heat gain (W) |
| φ | simultaneous factor |
| ψ | sensible heat proportion of the connection load |
| δ | spreading angle of the plume profile |
| λ | ratio between temperature and velocity profile widths |
| ρ | density of air in the plume (kg/m ³) |
| η | efficiency |

Subscripts

| | |
|-------|-------------------------------------|
| a | air |
| ads | air distribution effect on spillage |
| c | core/maximum |
| con | convective |
| esc | escaped air |
| exh | exhaust air |

| | |
|------------|----------------------|
| <i>h</i> | hydraulic |
| <i>o</i> | outdoors/surrounding |
| <i>occ</i> | occupied zone |
| <i>p</i> | pollution |
| <i>r</i> | room |
| <i>sup</i> | supply |
| <i>t</i> | temperature |
| <i>v</i> | velocity |

1. INTRODUCTION

Concerns over the indoor environment have increased during recent years as a result of knowledge about the significance of thermal conditions and air quality on health, comfort and productivity of the workforce. In a commercial kitchen, working conditions are especially demanding. There are four main factors affecting the thermal comfort, these being: air temperature, thermal radiation, air velocity and air humidity. At the same time, high emission rates of contaminants are released from the cooking process. Ventilation plays an important role in providing comfortable and productive working conditions and in securing the contaminant removal.

The published studies demonstrate quite clearly the health risk of cooking. Thiebaud (1995) indicates that the fumes generated by frying pork and beef are mutagenic. Hence, the chefs are exposed to relatively high levels of airborne mutagens and carcinogens. Shuguang (1994) analyzed various samples of cooking oil fumes. They found out that there is high concentration of carcinogens in cooking oil fumes. This is likely the reason for high incidence of pulmonary adenocarcinoma in Chinese women.

Vainiotalo (1993) carried out measurements at eight workplaces. His survey confirmed that cooking fumes contain hazardous components. It also indicated that kitchen workers may be exposed to relatively high concentration of airborne impurities. Li (1993) measured particle size distribution from scrambling eggs, frying chicken and cooking soup. Based on measurements, the concentrations of sub-micrometer particles increased 10-times during cooking.

Seow (2000) results show that inhalation of carcinogens generated during frying of meat may increase the risk of lung cancer among smokers. The risk was further increased among women stir-frying meat daily whose kitchens were filled with oily fumes during cooking. Ko (1997) reported that the risk of contracting cancer for non-smoking women appears to be associated with the efficiency of the fume extractor.

Although cigarette smoking is considered to be the most important cause of lung cancer, smoking behaviour cannot fully explain the epidemiological characteristics of lung cancer among Asian women, who rarely smoke but contract lung cancer relatively often. Ng (1993) found that over 97 % of the women in Singapore do not smoke. Thus, the presumable sources of indoor air pollution for housewives is passive smoking and cooking. This study indicates that greater relative odds of respiratory symptoms were associated with the weekly frequency of gas cooking. A statistical link between chronic cough, phlegm and breathlessness on exertion was also found.

Notami (1993) analyzed the association between occupation and cancer of lung and bladder in a case-control study in India. In this study, a statistically significant link between the lung cancers of cooks was found. For lung cases, comparing 'ever' employed with 'never' employed in a particular occupation, significantly elevated risks (adjusted for smoking) were found for cooks.

The previous studies highlight the importance of well-designed ventilation in the kitchen. The removal efficiency of the total system must be guaranteed and the spread of impurities throughout the kitchen should be prevented.

It is still quite common practice to estimate exhaust airflow rates based on rough methods. The characteristic feature of these methods is that the actual heat gain of the kitchen appliance is neglected. Thus, the exhaust airflow rate is the same: even if under the hood there is a heavy load like a wok, or light load like a pressure cooker. These kinds of rough estimation methods do not deliver optimal solutions. The size of the whole system will be oversized and so the investment costs and running costs will increase.

There are many methods available to determine the required exhaust airflow rate. For example face velocity (CP 13 2000), where airflow rate is determined by selected capture velocity and the area of the kitchen appliance under the hood. This method does not take into account the real heat gain of appliances. Hence in many cases, the estimation always exceed the actual requirement or demands Based on the case-study calculations, in the medium-load case rough method like CP 13 oversized the whole system 2- or 3- times compared with the actual demand. This over estimation is 1.4 –1.8 even with the extra-heavy load (Kosonen 2001).

A room energy balance approach is used in the previous VDI (VDI 1984). Based on the sensible load, the requested airflow rate is calculated (Eq.1). In the calculation, a special factor which takes into account the hood efficiency is also employed. The hood efficiency is not unambiguously explicated because the ratio of convection load, temperature gradient in the room space and the exact amount of the general exhaust are not determined.

$$q_{v,exh} = \frac{\varphi \cdot \sum_j (P_j \cdot \psi_j \cdot \eta_j)}{\rho \cdot c_p \cdot (t_r - t_{sup})} \quad (1)$$

where:

$q_{v,exh}$ = supply airflow rate, m³/s

φ = simultaneous factor of kitchen equipment

P_j = connected load of the kitchen equipment j

ψ_j = sensible heat proportion of the connected load of the equipment j, W/W

η_j = room load factor of the hood for the equipment j

ρ = density of supply air

c_p = specific heat capacity of air

t_r = room air temperature

t_{sup} = supply air temperature

A default value of η_j is set to 0.8 if at least 80 % of the kitchen exhaust air is removed via hoods.

It should be noted that with a hood it is only possible to capture the convection part of the load. Radiation will always be present in the room space. So, therefore the actual capture efficiency is only related to the convective part of the load.

A more accurate method is based on the heat gain of the appliances (VDI 1999). In this method, a consideration is made for the convective heat output, the area of the appliance and the distance between hood and appliance. There is also a factor for the supply configuration.

For example, using a low velocity supply solution leads to a lower extract airflow rate than with traditional mixing ventilation. The amount of air carried in a convective plume over a cooking appliance block at a certain height is calculated using Eq. 2:

$$q_{v,p} = k \cdot (z + 1.7D_h)^{5/3} \cdot (\Phi_c \cdot \varphi)^{1/3} \cdot r \quad (2)$$

Where

$q_{v,p}$ = airflow in convective plume, m³/h

z = height above cooking surface, m

φ = simultaneous factor

Φ_c = convective heat output of the cooking appliance block, W

k = empirical coefficient, k is 18 for a generic hood

r = reduction factor of installation place (free $r = 1$, near wall $r = 0.63$ or in the corner $r = 0.43$)

D_h = hydraulic diameter, m

$$D_h = \frac{2L \cdot W}{L + W} \quad (3)$$

L, W = length and width of the cooking surface accordingly, m

Since it is rare that all the equipment in the kitchen is operating simultaneously, the heat gain from cooking appliances is multiplied by the reduction factor called the simultaneous coefficient (φ). Normally, the simultaneous factor is from 0.5 – 0.8. This means that only 50 – 80 % of the appliances are used at any one time.

Kitchen hoods are designed to capture the convective portion of heat emitted by cooking appliances. Thus, the hood exhaust airflow should be equal or higher than the airflow in the convective plume generated by the appliance. The total of this exhaust depends on the general ventilation (Eq. 4)

$$q_{v,exp} = q_{v,p} \cdot K_{ads} \quad (4)$$

Where

K_{ads} = spillage coefficient taking into account the effect of the air distribution system.

The recommended values for K_{ads} (VDI 1999) are listed in the Table 1. Based on Table 1, the required exhaust airflow rate with the wall-mounted supply ($K_{ads} = 1.25$) is 19 % higher than with the low velocity diffusers ($K_{ads} = 1.05$) in the working area.

Table 1. Spillage Coefficients as a Function of the Air Distribution system

| Type of Air Distribution System | | K_{ads} |
|---------------------------------|--------------------------|-----------|
| Mixed flow | Tangential Air Opening | 1.25 |
| Mixed flow | Ceiling Opening | 1.20 |
| Laminar flow | Ceiling Air Opening | 1.1 |
| Laminar flow | Outlets in the work area | 1.05 |

Some codes (e.g. AS 1668.2 2002) use either a prescriptive or an engineered procedure for hood design. The engineered procedure is a performance-based approach that allows the utilisation of suitable technology to reach set targets. The solutions should be reviewed in the field or proven with appropriate calculations.

An other complementary method with the calculation method is to use measurements to analyse adequate airflow rate in the test conditions. The most popular methods are UL-710 (Underwriters Laboratories 1995) and F 1704- 99 (ASTM 1999). Both of these methods are based on visual observation. In UL, an inspector observes the capture and containment efficiency in the laboratory and based on visual test the minimum required airflow rate is fixed. In ASTM test, Schlieren technology is utilized to determine the threshold of capture and containment of a hood and appliance combination under idle and cooking conditions. The ASTM test gives an accurate platform for studying hood capture efficiency in different cooking processes.

None of these described calculation methods or measurement technologies are specially tailored for a kitchen ceiling environment. In normal design practice, empirical knowledge of the existing installations together with heat load based calculation have been used for airflow rate determination.

The ventilated ceiling approach offers a flexible solution for kitchens where the heat loads are relatively low and aesthetics are a concern (DW/171 1999). With the ventilated ceiling, it is possible to maintain good thermal conditions in the occupied zone with reasonable air flow rate (Akimoto et al.2002 and Horikawa et al. 2002). Structurally, the system consists of a stainless steel element that covers either the entire ceiling or only the active cooking area of a kitchen. It incorporates the air inlets, exhaust air outlets (including grease filters), and light fittings.

Ceilings are categorised as open and closed systems. In the open ceiling, air ductworks are connected to the voids above the ceiling. The open ceiling is usually assembled from supply and exhaust cassettes. The space between ceiling and void is used as a plenum. The more common closed type of plenum system comprises separate dedicated ductwork with connections to both supply and extract modules in the ceiling to avoid any risk of grease build-up in the void.

The efficiency of the exhaust system can be improved with a small capture jet installed at the ceiling surface. The air jet is projected horizontally across the ceiling, which helps to direct heat and air impurities towards the exhaust. This capture jet represents only about 10 % of the total supply air flow rate.

Although the use of hoods is ideal for handling contaminants produced in concentrated areas, the use of the air conditioning ceiling should be considered as a viable option. They are particularly suitable for the following applications (DW/171 1999):

- Structural limitations (e.g. a low ceiling level makes the use of hoods impractical;
- False ceiling aesthetics are important and visibility cannot be impaired by hoods;
- The cooking equipment does not generate intensive output in concentrated areas;
- A good level of extraction is required but the level of contaminant produced is relatively low.

The focus segment for the research was commercial kitchen ventilation. The main target was to get improved basis for designing a ventilated ceiling system and a better knowledge of the effect of thermal plumes and supply air systems on the performance and efficiency of a ventilated ceiling. The objectives of this study have been to investigate the actual thermal plumes of kitchen appliances during idle and cooking modes, capture efficiency of a ventilated ceiling and the effect of supply air systems on the efficiency of a ventilated ceiling.

The specific objectives for the research were:

- To determine the actual plume characteristics of typical kitchen appliances during idle mode (I)
- To validate the accuracy of the generic plume equation during cooking mode (II)
- To establish the correlation between convection load and the maximum excess temperature, maximum velocity and the momentum flux (III)
- To evaluate the effect of a capture jet on the contaminant removal efficiency of a ventilated ceiling (IV)
- To develop a model for the capture and containment efficiency analysis of a ventilated ceiling and to compare the efficiency with and without the capture jet concept (V)
- To derive the flush-out factor of the supply air and to study the effect of different supply air systems on the efficiency of a ventilated ceiling (VI).

2. RESEARCH METHODS

Laboratory measurements of thermal plumes, tracer gas measurements of ventilation efficiency and computational fluid dynamics simulations of the mock-up kitchen were conducted in order to study the actual plumes of typical kitchen appliances, the capture efficiency of a ventilated ceiling and the effect of the supply system on the efficiency of a ventilated ceiling. These methods and experimental set-ups are described in this chapter.

2.1 Physical Experiments

2.1.1 Thermal Plumes of Kitchen Appliances

The thinking behind these measurements was to analyze the convection flows of the actual kitchen appliances during the idle and cooking modes. The measurements were conducted in the laboratory of the Finnish Institute of Occupational Health.

The measurements were carried out in a test room built inside a laboratory facility. Construction consists of a steel frame, with floor dimensions of 10 m x 4 m and a height of 6 m. The room space is thermally insulated from the surroundings with 50 mm thick polystyrene elements. During measurements, the temperature difference between the average wall temperature of the test room and ambient air temperature at 1.1 m level was less than 2.0 °C. The vertical temperature gradient in the room space varied between 0.2 – 1.1 °C per m depending on the tested appliance.

The supply air flow rate was delivered, using the displacement ventilation principle, at floor level from six multi-nozzle ductworks, guaranteeing an undisturbed convection flow from a kitchen appliance, Fig. 1. The total supply air flow rate of 600 l/s was adjusted to cover the induced air flow rate of the convection flow above the 3 meters level. The supply temperature was about 20 °C. The return air grille was installed at the height of 6 m.

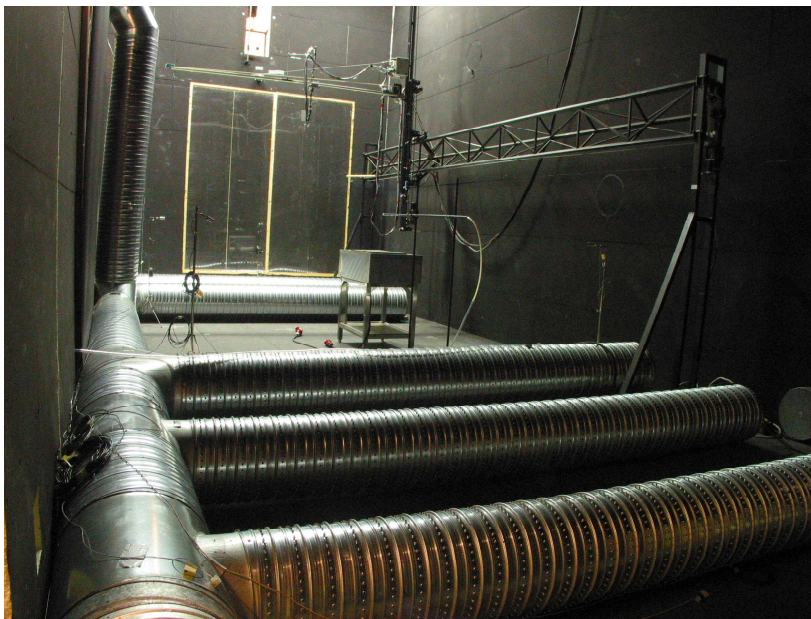


Figure 1. The installation principle of a kitchen appliance and supply air devices.

Convection plumes from an iron range, a chrome range, a gas range and an induction griddle were studied under idling and cooking conditions. In addition, an induction range and a fryer were studied during the cooking mode. The cooking mode of the ranges was facilitated by boiling water in two 10 L kettles, Fig. 2 In the case of the induction griddle, cooking was executed by frying boneless chicken breast, Fig. 3. In the fryer, the oil temperature was maintained at 180 °C.

The installation principle is different from that used in laboratory studies of kitchen appliances, where the volume flow rate of thermal plume is measured together with a hood in a wall-type installation (Gerstler 1999). It should be noted that the island-type installation approach without any local exhaust, utilized in this study, makes it possible to get a generic view of the plume distribution.

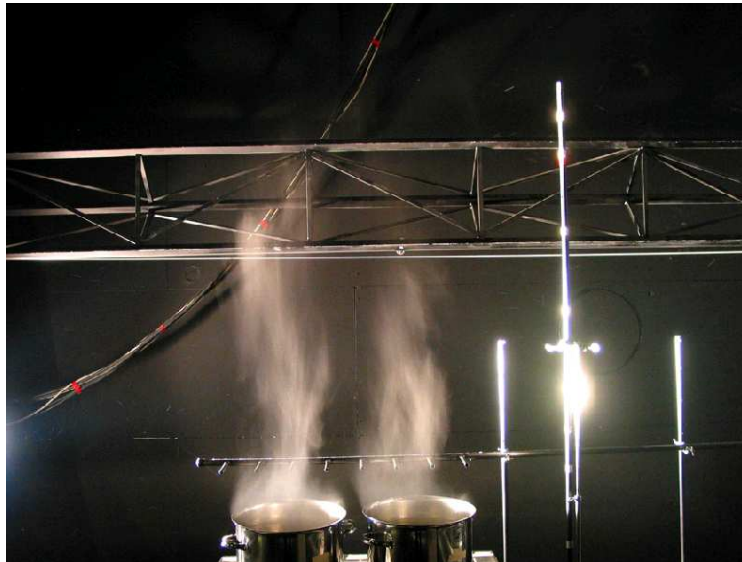


Figure 2. The cooking arrangement of the ranges.

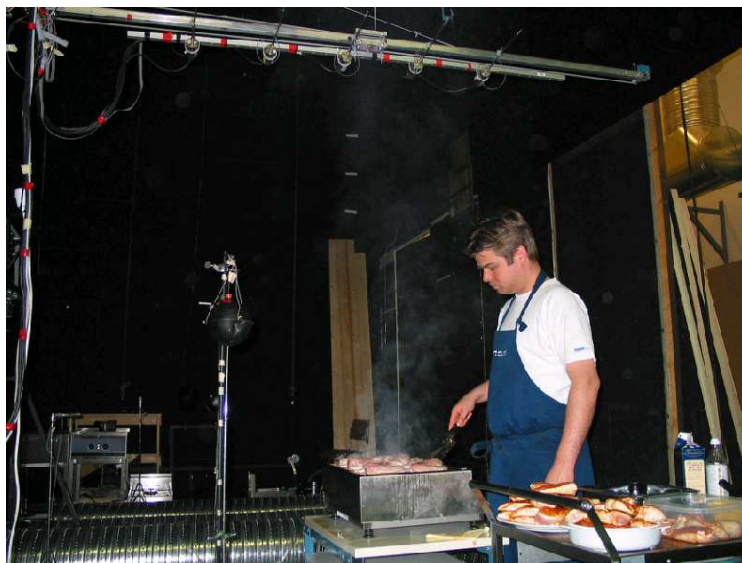


Figure 3. The cooking arrangement of the induction griddle.

The studied appliances represent the state-of-the art technology of kitchen appliances. The iron range is a typical range used in commercial kitchens. The chrome range has a temperature sensor in the surface plate that improves energy efficiency during partial load conditions. Furthermore, the surface is a low-emission material that reduces radiation compared with the traditional cast iron surface. The induction griddle has fast and accurate control that maintains a constant surface temperature during different cooking conditions.

During the tests, the actual power of electric appliances was measured with a clip-on-ammeter. The power of the gas range was determined through the consumed quantity of gas, Fig. 4.

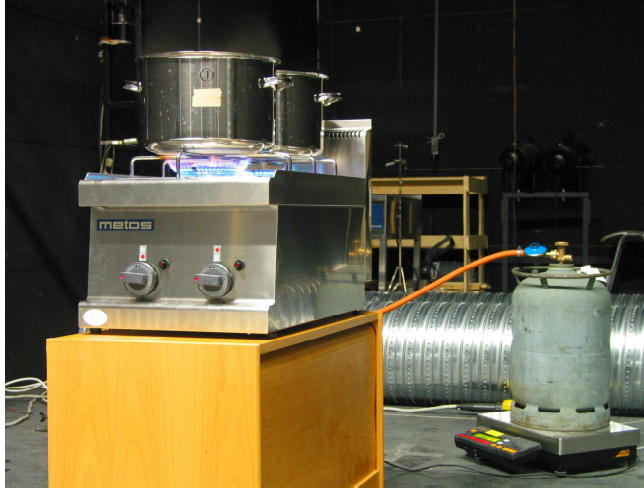


Figure 4. The measurement arrangement of the power of a gas range.

It should be noted that the power of kitchen appliances can also be determined from the energy balance in the test room as described by ASTM (1996). However, the direct power measurement was selected in this study because of the simplicity and accuracy of the measurement procedure. The nominal connection powers and the dimensions of the appliances are presented in Table 2.

Table 2. Description of the measured kitchen appliances.

| Appliance and connection Power | Description |
|--------------------------------|---|
| Iron Range 6 kW | Two burners range surface material: cast iron 500x800x950 (H) |
| Chrome Range 6 kW | Two burners range with control of the surface temperature. Surface material: chrome 500x800x950 (H) |
| Gas Range 5.5 kW | Two burners (2 kW and 3.5 kW) range 400x650x460 (H) |
| Induction Range 10 kW | Two burners with demand based power control 380x700x145 (H) |
| Induction Griddle 6 kW | Electronic power control. Surface material: stainless steel 520x440x175 (H) |
| Fryer 6.9 kW | 360x435x 260 (H) |

The velocity and temperature measurements were performed using a measurement robot, Fig. 5. The convection load was determined based on temperature and velocity measurements on a horizontal plane. The probes were attached to a computer-controlled traversing system moving them from point to point and scanning the determined four measurement planes at the heights of 0.8 m, 1.2 m, 1.6 m and 2.0 m from the appliances.

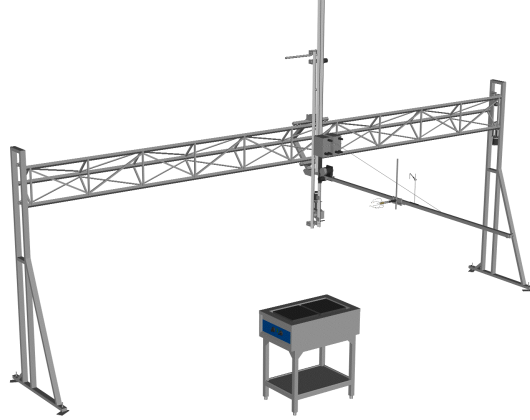


Figure 5. The used traversing system

The basic measurement grid of 1.1 m x 1.1 m consisted altogether of 121 measurement points (0.1 m interval) in each plane. In Figure 6, the basic measurement grid and an example of the velocity distribution are presented. The boundary of the thermal plume was established visually taking into account the actual Gaussian velocity distribution in the cross section of the maximum air velocity. Outside of the determined boundary velocity (between 0.01-0.04 m/s), the background velocity was computed as an average velocity of the measurement points. When the air flow rate was computed, that background velocity was then subtracted from the measured vertical velocity component.

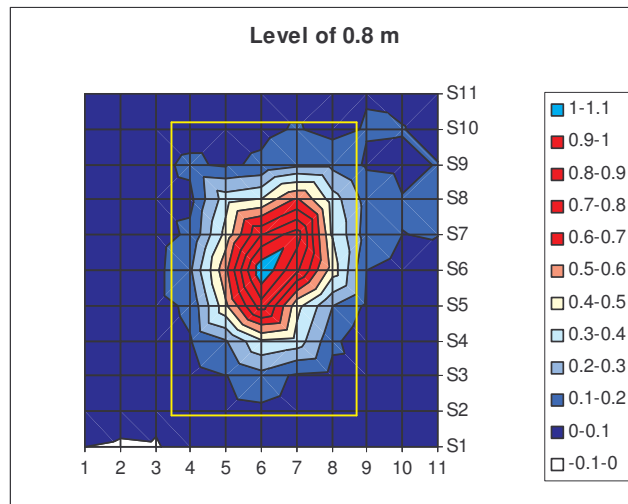


Figure 6. The measurement grid (10 x 10 cm). The velocity distribution of an iron range (50 x 80 cm) at the height of 0.8 m during idle mode. The centre point of a range is at 6 & S6.

Accurate measurement of air velocities with high turbulence intensities is difficult. In turbulent flows, the velocity changes its magnitude and direction. Rapid velocity changes are not detected by a probe with a long response time. Melikov (Melikov and Sawachi 1992) has pointed out that a response should be 0.2 –0.5 s. In previous publications, hot wire, vane and

hot sphere anemometers, smoke, video recording and digital image time processing procedures as well as laser-Doppler- anemometers, have been used to measure the velocity distribution in convection flows (e.g. Kofoed (1991), Mundt (1996) and Welling (1993).

In this study, the air velocity was measured with Kaijo Denki WA- 390 ultrasonic probes, which have an accuracy of 0.02 m/s for each velocity component. The sensors measure air velocity vector components with three pairs of ultrasonic transducers by registering the flight time of an ultrasonic pulse. The response time of the sensor is 0.5 s. The air temperatures were measured with Fenwal thermistors with an accuracy of 0.1 K. Also, the relative humidity was measured with Vaisala HMP 143 sensor. In the traversing system, the measurement sensors were connected together, Fig. 7. The installation distance of the velocity and temperature measurement probes (about 10 cm) was taken into account when the convection heat gain was determined.

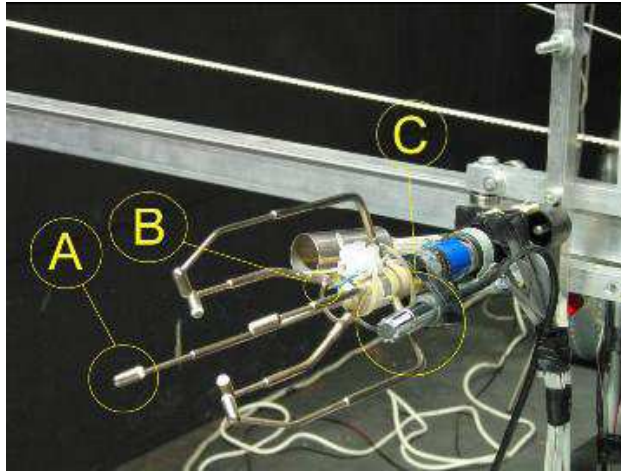


Figure 7. The measurement arrangement of probes. A) one transmitter-receiver of ultrasonic probe, B) temperature sensor and C) relative humidity sensor

The sensor output was sampled by a data logger with a 1 s sampling time. Measurements of air velocity and temperatures were time averages over 60 s and the values were recorded with a data logger. In each measurement sequence when the traversing system moves the sensors, there was set a delay time of 10 s to eliminate possible mechanical fluctuation of the traversing system. The radiation effect of the iron range on the temperature probes was corrected by separate reference measurements.

In the case of the induction griddle, the ultrasonic probes could not be used for the air velocity measurements because of the contaminants released in the frying process. In this case, the more robust omnidirectional TSI 1620 hot sphere velocity probes were used instead, Fig. 8.

The output of these sensors was later compared with the ultrasonic sensors by a separate plume measurement without the frying and a calibration curve was determined to correct the reading of the omnidirectional sensors. The requirement for this correction comes from different measurement principles of the two sensor types. The time-averaged reading of the omnidirectional sensors is affected by turbulent fluctuations, which must be corrected when calculating the flow rate through the measurement plane, Fig 9. The averaging time was also reduced in the induction griddle measurement to 30 s because of the time constraints of the frying.

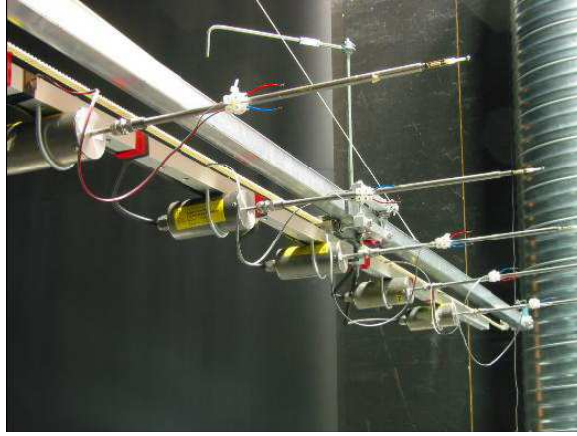


Figure 8. Hot sphere velocity probes were used in the frying process of an induction griddle.

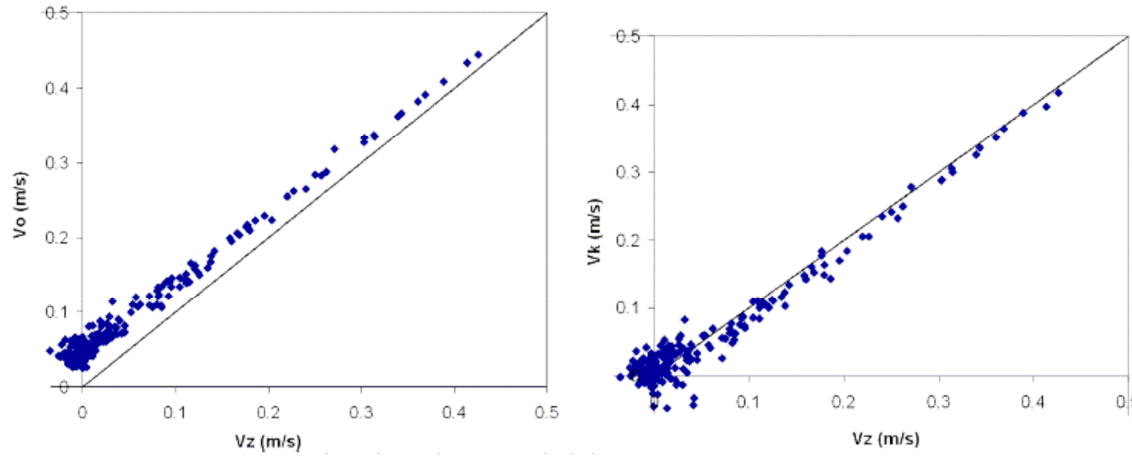


Figure 9. The reading of omnidirectional hot sphere velocity probes (v_o) (left panel) and the determined calibration reading (v_k) (right panel) as a function of the reading of the ultrasonic probe (v_z).

2.1.2 Trace Gas Measurements of a Mock-Up Kitchen

The principal idea of the measurements is to analyze the effect of different capture jet concepts on the efficiency of a ventilated ceiling. With laboratory measurements, the containment removal efficiency of a ventilated ceiling was evaluated.

The measurements were performed in laboratory conditions with a mock-up kitchen at Halton's facilities. In the case-study kitchen, the kitchen appliance block consists of three electric appliances: 1) a griddle, 2) an iron range and 3) a combi (combination of a hot plate and a griddle). The studied appliance block represents the state-of-the art technology of kitchen appliances. During the part-load conditions tests, the actual power of the electric appliances was measured with a clip-on-ammeter. The total heat load of the appliance block

was 7.9 kW. The description of the appliances, the temperatures of the cooking area and the electric loads are presented in Table 3.

Table 3. Description of the kitchen appliances in the measured appliance block.

| Appliance | Dimension | Power Rating (kW) | Actual Capacity (kW) | Surface Temperature (°C) |
|-------------------------------|------------------------------|-------------------|----------------------|--------------------------|
| Griddle | 800 x 900 x 920 (H) | 11.1 | 4.0 | 170 |
| Iron Range | 400 x 900 x 900 (H) | 10.2 | 2.6 | 300 |
| Combi (hot plate and griddle) | 1300 x 800 x 950 (H) | 11.2 | 1.3 | 260 185 |
| The whole appliance block | 2500 x 800-900 x 900-950 (H) | 32.5 | 7.9 | - |

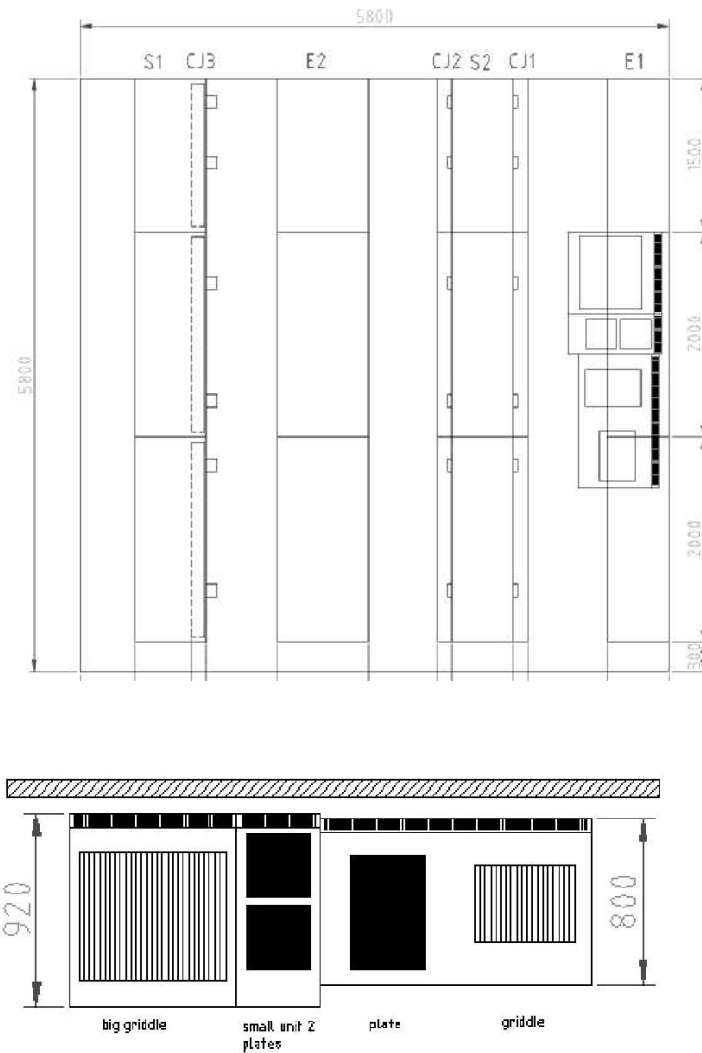
The studied ventilated ceiling system consists of stainless steel elements that cover the entire ceiling area. The ceiling comprised exhaust, supply and capture jet units, with lights and ceiling elements between the exhaust and supply units. The capture jet air is supplied horizontally across the ceiling. This jet helps to direct heat and air impurities towards the exhaust.

The measurements were carried out in a case-study kitchen room with the ventilated ceiling dimension of 5.8 m x 5.8 m and a height of 2.6 m. In the case-study kitchen, there are two exhaust (E1 and E2) and two supply units (S1 and S2) equipped with the capture jet units (CJ1-CJ3), Fig. 10.

In the test kitchen, also installed is a corner-type floor mounted low velocity unit. Depending on the ventilation strategy employed, the appropriate supply and exhaust modules were enabled. In most of the conducted tests, the kitchen appliance block was installed close to the wall under the exhaust unit E1. In one test case, an island-type of installation was used and the same appliance block was installed underneath the exhaust unit E2.

The effect of the selected ventilation concept on the contaminant removal efficiency was studied with a tracer gas method. In this study, the constant dosing method was employed.

The gas analyzer together with the sampler and doser unit and a controller computer makes it possible to perform multi-point monitoring and dosing tasks. The tracer gas used in this study was SF₆. During the test, SF₆ concentrations were sampled at 10-min intervals and the total length of the tests varied between 1.5 – 2.5 hours to ensure representative conditions. For the dosing system, the volume flow rate of the tracer gas supply was adjusted to give reasonable concentrations. The dosing point was installed just over the hot plate of the combi that compounds effectively tracer gas in the convection flow.



B1 (B2) COOKING EQUIPMENT

Figure 10. The layout of the case-study kitchen and the locations of the exhaust (E1 and E2), supply (S1 and S2) and the capture jet units (CJ1-CJ3). The supply and exhaust units are built with two 2.0 m and one 1.5 m modules.

The sampling system has 6 inlet channels and one dosing channel. Four sampling points (P3-P6) are located close to the breathing zone at 1.6 m level from the floor in the occupied zone. In addition, there are sampling points in the exhaust and supply ductwork. The location of the dosing and sampling points are presented in Fig. 11.

It should be noted that a minor part of the tracer gas was re-circulated back to the supply air because the exhaust air of the laboratory facilities is released in the same factory hall where the supply air is taken. The effect of the tracer gas concentration in the supply side was taken into account when the containment removal efficiency was calculated.

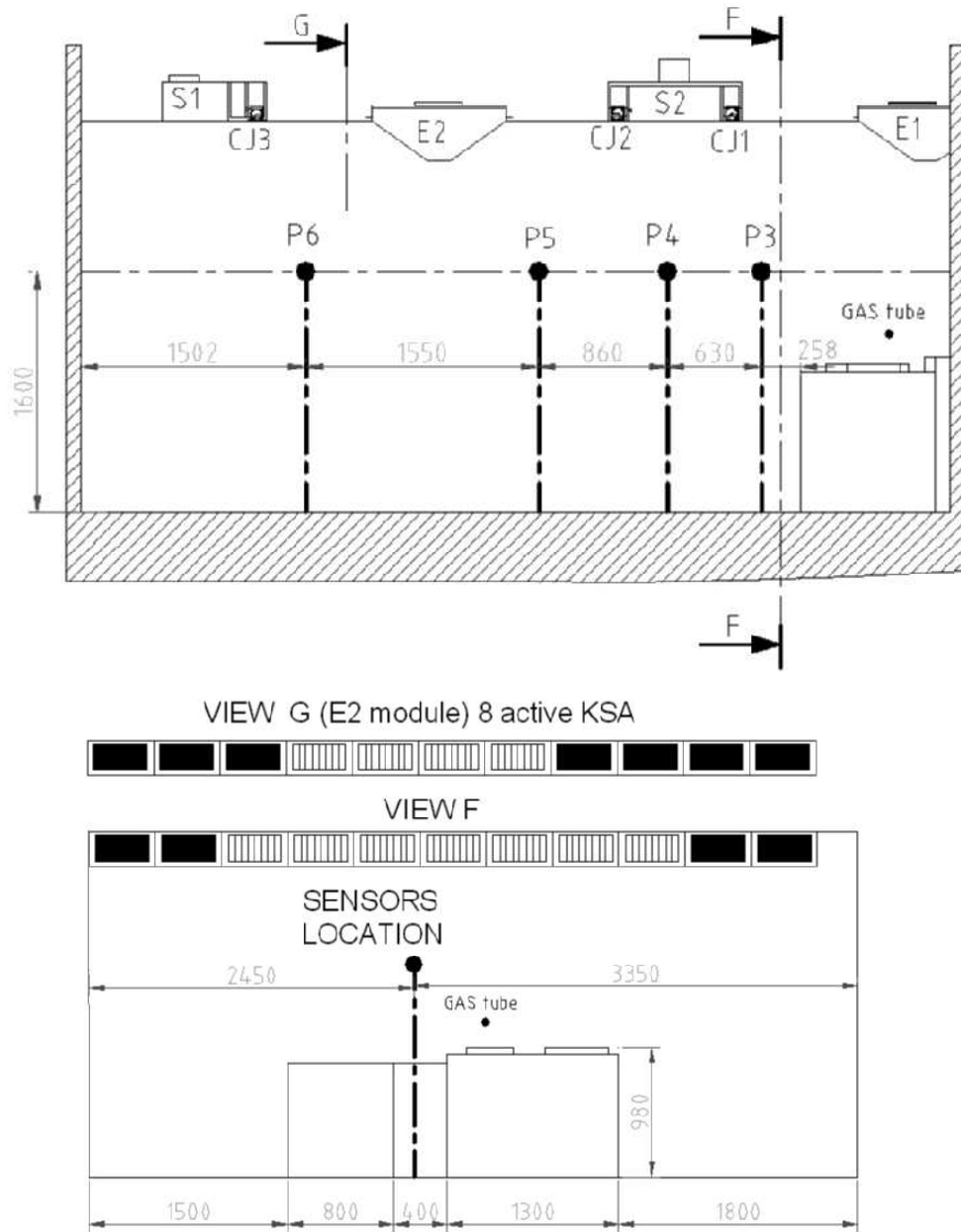


Figure 11. The locations of the tracer gas dosing and sampling points

In the study, there were two basic capture jet concepts: 1) full-length and 2) centralized exhaust and capture jet. As a reference system, the thermal displacement is studied without the capture jet. In the tests, the exhaust and supply air flow rates are varied between 580 - 1,200 l/s. Infiltration from the surrounding space was controlled by balancing the exhaust and supply air flow rates. In all tests, the capture jet was fixed at 10 % of the exhaust air flow rate. Fig.12 presents the studied capture jet concepts, the air flow rates of the total modules and the active parts of the exhaust and capture jet modules.

In the full-length exhaust and capture jet concepts, the total length of the supply and exhaust modules were utilized. In the centralized concepts, the active parts were situated mainly over the appliance block. The effect of the location of supply unit was also analyzed by introducing the supply air close to the opposite wall using the centralized concept.

The supply and exhaust units were built with two 2.0 m and one 1.5 m modules. In the exhaust modules, there were three (1.5 m) or four (2.0 m) length of 500 mm grease filters. In the wall type of the kitchen appliance installation, tests were conducted with 11 (the full-length concept) and 7 (centralized concept) active filters in exhaust unit E1. In the island type of the kitchen appliance installation, there were 4 active filters in both sides of the central exhaust module E2 i.e. 8 activated filters in total (Fig.12).

In a separate test, the impact on efficiency of activating two exhaust and supply units at the same time was also studied. In all the previous tests, the kitchen appliance block was locating close the wall. In one test, the appliance block was an island-type installation and both supply units (S1 and S2) were employed together with the exhaust unit (E2).

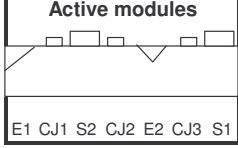




| CASES | Air Flow Rate l/s | <div>Active modules</div>  | Active parts in each modules and air flow rates l/s | | | | | | | | | | | | | | | Displacement Unit |
|---------------------|-------------------------|--|---|-----|------|---|------|---|-------------|----------|----------|-------------|---------|---------|--|---|-----|----------------------|
| | | | Exhaust unit | | | | | | Capture Jet | | | Supply unit | | | | | | |
| | | | E 1,5m | | E 2m | | E 2m | | CJ 1,5m | CJ 2m | CJ 2m | S 1.5m | S 2m | S 2m | | | | |
| | | | 1 | 2 | 3 | 1 | 2 | 3 | | | | | | | | 4 | | |
| Full-length | | | | | | | | | | | | | | | | | | |
| Case 1/W | 580 |  | 158 | 211 | 211 | | | | 16 | 21 | 21 | 142 | 190 | 190 | | | | |
| Case 2/W | 700 |  | 191 | 255 | 255 | | | | 19 | 25 | 25 | 172 | 230 | 230 | | | | |
| Case 3/W | 850 |  | 232 | 309 | 309 | | | | 23 | 31 | 31 | 209 | 278 | 278 | | | | |
| Centralized | | | | | | | | | | | | | | | | | | |
| Case 4/W | 700 |  | | | 700 | | | | | 70 | | 172 | 230 | 230 | | | | |
| Case 5/I | 700 | | | | 700 | | | | | 35 | | | 315 | | | | | |
| | | | | | | | | | | 35 | | | 315 | | | | | |
| Case 6/W | 850 | | | | 850 | | | | | 85 | | 208 | 278 | 278 | | | | |
| Other Supply | | | | | | | | | | | | | | | | | | |
| Case 7/W | 850 | | | | 850 | | | | | | | 208 | 278 | 278 | | | | |
| Case 8/W | 850 | | | | 850 | | | | | | | | | | | | 850 | |
| Case 9/W | 1200 | | | | 700 | | | | | 70 | | 172 | 230 | 230 | | | | |
| | | | | | 500 | | | | | 50 | | 123 | 164 | 164 | | | | |

Figure 12. The description of the supply air cases.

2.2 Computational Fluid Dynamics Simulations

The effect of a capture jet on the efficiency of a ventilated ceiling was evaluated using CFD-simulations. All computations were performed using AirPak 2.0.6. The equations for conservation of mass, momentum and energy were solved using the finite-volume method.

Computational fluid dynamics refers to two independent disciplines. Fluid dynamics considers the selection of governing equations for the solution of a given problem, including boundary conditions, turbulence models, empirical relations, and physical properties. According to Roache (1997) this is about the solving the right equations. Computational on the other hand refers to the selection of methods required to solve the chosen set of equations, including discretization schemes, computational grids, and convergence grids. Sorensen and

Nielsen (2003) have classified and discussed the major sources of errors in indoor environmental investigations using CFD. These errors are divided into two main groups: 1) poor choice of the governing equations and 2) poor solution of the governing equations.

In this thesis, the standard k - ε model was employed for a turbulence closure and ideal gas law in modeling buoyancy, respectively. The standard k - ε model was chosen to model turbulence because it represents the well-known and validated model for air distribution analysis.

In turbulent flow, the variables can be divided into one time-averaged part and one fluctuating part. The model is based on a transport equation for turbulent kinetic energy k and a transport equation for the dissipation of turbulent kinetic energy ε (Launder and Spalding, 1974). In the derivation of the standard k - ε model, it is assumed that the flow is fully turbulent, and the effects of molecular viscosity are negligible. The standard k - ε model is therefore exactly valid only for fully turbulent flow.

The radiation heat transfer between heat source and room surfaces was calculated by utilizing AirPak's surface-to-surface blackbody assumption model. AirPak calculated the view factors applying between the source and surfaces.

The wall treatment was achieved by use of the standard wall function provided by the software. In all the cases the segregated solver has used to obtain results for flow, turbulence and energy throughout the computational domain. All simulations were performed as steady-state.

First-order upwind scheme discretization of governing equations was used, because in this application it gives much better convergence than the second-order scheme. Body-force-weighted scheme was used for discretization of pressure. The scheme is suitable for high-Ryaleigh-number natural convection flows. The pressure-velocity coupling was achieved by SIMPLE algorithm. Specific under-relaxation factors for the segregated solver were modified to obtain converged results.

The room was divided into a hexahedral grid system with about 325,000 cells and a local refining mesh in critical regions. Based on grid refinement studies, this mesh was deemed to be sufficiently fine to capture all significant flow features and the concentration distribution. Using a finite volume method, the calculation domain in Fig. 13 is divided into a finite number of control volumes and grid points. The current research used only the structural hexahedral mesh depicted in Fig. 13. It is important to generate high-quality grid layouts near the supply units and the heat sources (Chen and Srebric 2001). In this study, the grid layout near the supply units and the kitchen appliance was more dense to improve the accuracy of the calculation.

The commercial CFD codes normally have a default convergence criterion and when the residual is below that criterion, convergence of the solution is assumed. In this thesis, the convergence criteria used ($1e-4$) is based on the recommendation of AirPak.

Commonly used boundary conditions, i.e. an inlet boundary and a pressure boundary, were applied at the air inlet and outlet of the ventilation system. The box method is applied when the horizontal capture jet is modelled. In the box method (Nielsen 1992), the boundary

conditions are moved in the room space and described by wall jet profiles at the surface of the box surrounding the diffuser. The momentum modelling approach (Chen and Jiang 1992) is used when the vertical low velocity ceiling supply unit is modelled. In this method, the supply unit imposes initial jet momentum as a boundary condition for CFD simulations. The boundary condition of the exhaust opening has little impact on room airflow. However, it is an flow parameters were used as a boundary conditions of the exhaust openings. The air flow rate was distributed at a predetermined ratio through the outlets. Openings at the bottom of doors were used for infiltration air to the kitchen. In Fig. 13, the boundary conditions used in the simulated mock-up kitchen are shown.

The boundary conditions of heat gains are determined using the known heat flux of the kitchen appliance. In the calculation, the surface temperature of the surrounding walls were set to a constant 21 °C and the emission factors used were based on the surface material.

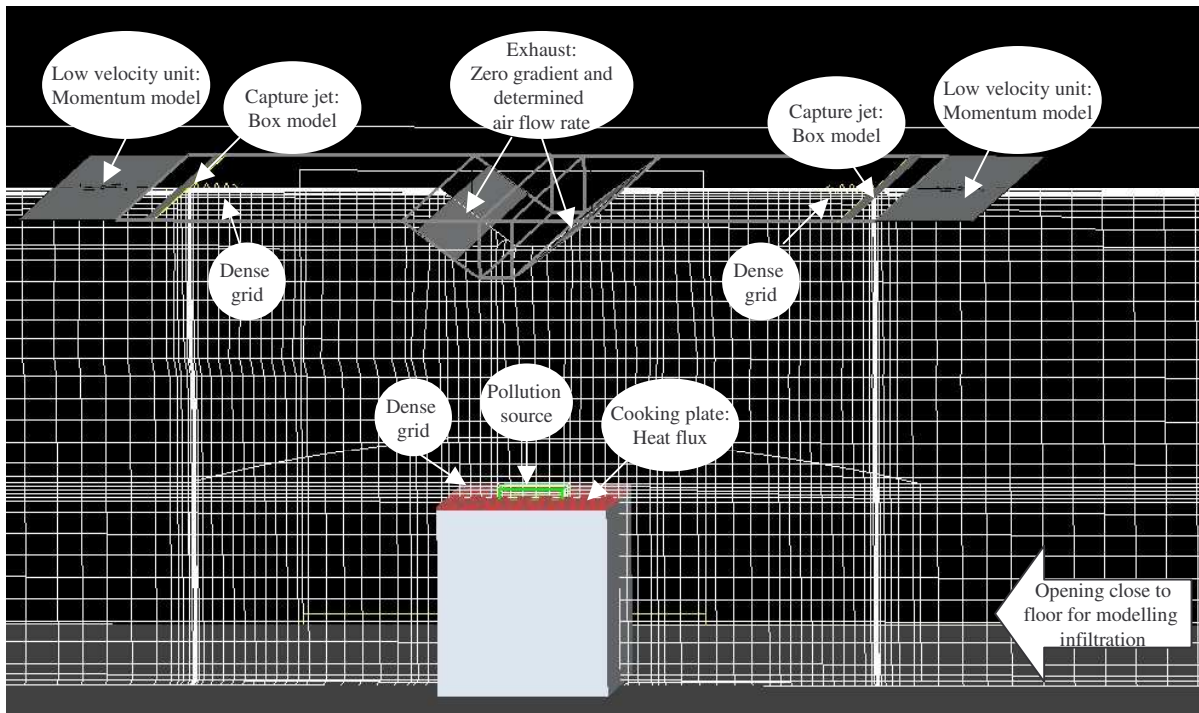


Figure 13. Grid layout in the simulated cases and the set boundary conditions

The kitchen appliance was modelled for heat gain. The pollution source in the simulations was 24.7 g/s of water vapour. It was modelled with the diffusion-convection equation to predict the local mass fraction of pollution. The mixture of the pollution source and air flow are calculated as a multi-component flow. The AirPak software calculates appropriate average values of the properties for each control volume in the flow domain. These average values will depend both on component property values and on the proportion on the air and water vapour in the control volume.

In the multi-component flow, it is assumed that the various components of a fluid are mixed at the molecular level, that they share the same velocity, pressure and temperature fields, and the mass transfer takes place by convection and diffusion. Using this model, the water vapour will affect the flow of the fluid through the fluid properties. However, the water vapour and air are assumed to have a common velocity.

This was supported by laboratory measurements undertaken in another study (Lappeenranta 1994). The laboratory measurements were conducted with and without the capture jet in a simple one-appliance-kitchen layout. The measurements were conducted in laboratory conditions with a mock-up kitchen at the Halton facilities. The floor area of the kitchen is 6.5 m x 9.5 m. The ventilated ceiling is 3.5 m x 4.3 m in area and is located either at 2.3 m or 2.6 m above floor level. Figure 14 shows the ventilated ceiling concept and the three measurement points underneath the structural ceiling.

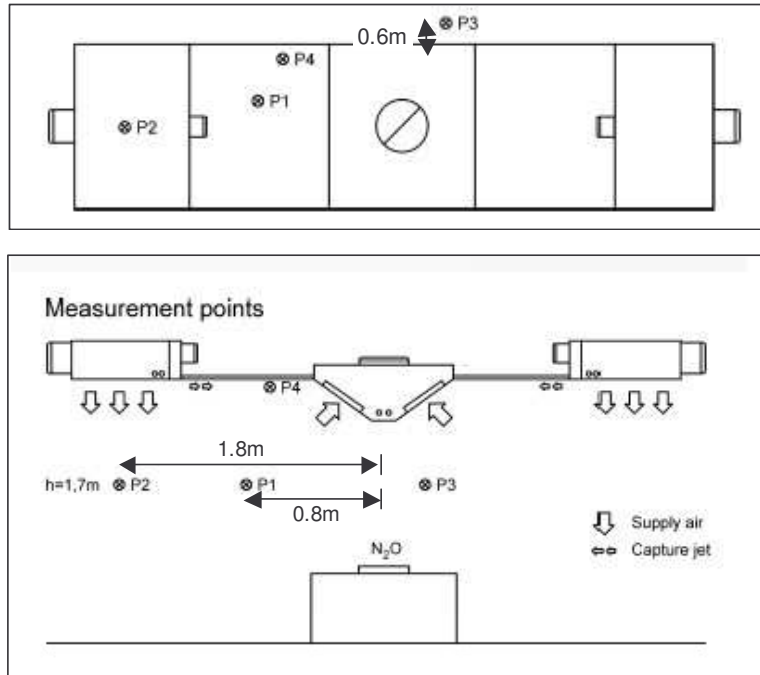


Figure 14. The layout of the mock-up kitchen with measuring locations (P1... P4).

The studied ventilated ceiling comprised exhaust, supply and capture jet units, with lights and ceiling elements between the exhaust and supply units. The capture jet air is supplied horizontally across the ceiling. This jet helps to direct heat and air impurities towards the exhaust. The kitchen appliance (size 1,200 mm x 800 mm x 870 mm (H)) consisted of a range and a frying pan. The surface temperature of the appliance was about 200 °C with a total heat gain of 5.6 kW. The supply air temperature was 18 °C and the room air temperature was 22 °C.

Measurements were conducted with and without the capture jet at three different air flow rates (100 %=design value, 150 % and 50 %). Table 4 presents the studied cases. It should be noted that in the cases of 50 % and 100 % air flow rate the room space was under-pressurized. In the CFD- analysis, the infiltration was modelled with the opening at the bottom of doors.

Table 4. The design principle and the air flow rates in the studied cases.

| Design principle | Air flow Rates (l/s) Supply+ Capture Jet / Exhaust |
|------------------|---|
| 50 % | 245+ 65 / 400 (with capture jet) |
| 50 % | 310+ 0 / 400 |
| 100 % | 670+120 / 840 (with capture jet) |
| 100 % | 790+ 0 / 840 |
| 150 % | 920+170 / 1090 (with capture jet) |
| 150 % | 1090+ 0 / 1090 |

Contaminant distribution was examined by releasing nitrous oxide N₂O tracer gas on the range with a constant flow rate of 210 l/h using a spiral spreader. The concentration of the tracer gas was measured at four locations in the kitchen (Fig. 14). Two sampling points were located underneath the ventilating ceiling in the occupied zone (P1 and P2 at 1.7 m level), and the third point (P3 at 1.7 m) was located outside the ventilating ceiling area. The fourth point (P4) was located near the edge of the ventilating ceiling 0.2 m from the grease extraction-unit.

3. THE THEORY OF THERMAL PLUMES

The most accurate method to compute the requested air flow rate in commercial kitchens is based on the heat gain of the appliances. Theoretically, the amount of air carried in a convective plume over a cooking appliance block at a certain height is possible to calculate using generic theory of thermal plumes.

The turbulent plume flows have been studied extensively for a great many years (Turner 1972). The flows studied were either convection thermal plumes (Popiolek 1981, Popiolek and Mierzwinski 1984, Mierzwinski 1992) or buoyant jets for which an initial momentum is imparted to the source (Papanicolaou and List 1988, Dai et al 1994). Assuming a self-similar mean behaviour, it is possible, based on a dimensional analysis (Batchelor 1954), to describe the behaviour of the fully developed turbulent plume by correlations involving the distance from the heat gain and the initial buoyancy flux.

Basic assumptions about a plume's axial velocity, excess temperature and plume width from a point heat source have been presented by Schmidt (1941) and Morton (1965). Schmidt makes basic assumptions about the flow from a point heat source. The axial velocity, excess temperature and the width are presented as a function of the vertical distance in Eqs. 5-7

$$v_c(z) = f(z^{-1/3}) \quad (5)$$

$$\Delta T(z) = f(z^{-5/3}) \quad (6)$$

$$r(z) = f(z) \quad (7)$$

Rouse et al (1952) introduced expressions for the volume flux, momentum flux and kinetic energy flux by using the similarity hypothesis. The following expression for the air flow rate q_v , momentum flux M and kinetic energy E are obtained:

$$q_v = 2 \cdot \pi \int_0^{\infty} v \cdot r \cdot dr \approx z^{5/3} \quad (8)$$

$$M = 2 \cdot \pi \int_0^{\infty} \rho \cdot v^2 \cdot r \cdot dr \approx z^{4/3} \quad (9)$$

$$E = 2 \cdot \pi \int_0^{\infty} \frac{1}{2} \cdot \rho \cdot v^3 \cdot r \cdot dr \approx z \quad (10)$$

The method to calculate the volume flow rate in plumes based on Gaussian form is used e.g. by Popiolek (1981). In that method, the velocity distribution were approximated by Gaussian distribution calculated from maximum v_c , width R_v , and distance r :

The mean velocity distribution can then be calculated from:

$$v(r) = v_c(z) \cdot e^{-\frac{r^2}{R_v^2}} \quad (11)$$

The temperature difference is calculated using the following equation, respectively

$$\Delta T(r) = \Delta T_a \cdot e^{-\frac{r^2}{R_t^2}} \quad (12)$$

where

ΔT_a = the maximum excess temperature

R_T = the width of the temperature profile

When approximating the results with the Gaussian distribution, it is possible to calculate the buoyant jet properties (Kofoed 1991).

The volume flow rate (q_v) can be calculated with the Equation 13:

$$q_v = \pi \cdot v_c \cdot R_v^2 \quad (13)$$

The vertical momentum flux, M:

$$M = \frac{\pi}{2} \cdot \rho \cdot v_c^2 \cdot R_v^2 \quad (14)$$

where

ρ = local plume density

The kinetic energy flux, E:

$$E = \frac{\pi}{3} \cdot \rho \cdot v_c^3 \cdot R_v^2 \quad (15)$$

The enthalpy flux, H:

$$H = \pi \cdot \rho \cdot c_p \cdot v_c \cdot \Delta T_0 \cdot \frac{R_v^2 \cdot R_T^2}{R_v^2 + R_T^2} \quad (16)$$

where

c_p = specific heat at constant pressure

Further, the local Archimedes' number Ar defined on a local velocity radius scale as well as the width ratio between the temperature excess and velocity profile (λ) can be calculated as:

$$Ar = \frac{\beta \cdot g \cdot \Delta T_0 \cdot R_v}{v_c^2} \quad (17)$$

$$\lambda = \frac{R_T}{R_v} \quad (18)$$

where

β = thermal expansion coefficient

g = acceleration due to gravity

Popiolek (1981) has analyzed the plume from a point heat source assuming Gaussian shaped profiles, and presents a local approximation by using a model of a plume above a point heat source. Expressions for the local Archimedes' number and the ratio factor, by means of the temperature excess and velocity distribution factors gives:

$$Ar = \frac{2 \cdot p}{3 \cdot m^{\frac{2}{3}}} \quad (19)$$

$$\lambda = \left(\frac{m}{p}\right)^{\frac{1}{2}} \quad (20)$$

The parameter m is called the velocity distribution factor and it describes the width of the velocity profile. It also characterizes the angle of spread. The parameter p is called the temperature distribution factor, having the same function for temperature.

If Eq. 18 is introduced, the temperature excess and velocity distribution factors, p and m can be calculated explicitly as:

$$p = \frac{4}{9} \cdot Ar^{-2} \cdot \lambda^{-6} \quad (21)$$

$$m = \frac{4}{9} \cdot Ar^{-2} \cdot \lambda^{-4} \quad (22)$$

The entrainment factor α describes how effectively the convection flow induces room air. The entrainment factor can be calculated according to Eq. 23.

$$\alpha = \frac{5}{6} \cdot m^{-\frac{1}{2}} \quad (23)$$

The theory of plumes is widely exploited e.g. in displacement ventilation design practice. For the practical application, analytical equations to calculate velocities, temperatures and air flow rate in a thermal flow above point and line sources are presented e.g. Mundt (1996) and Baturin (1972).

In practice, heat sources are seldom a point source. The most common approach to account for the real source dimensions is to use a virtual origin from which the air flow rate are calculated, Eq. 24.

$$q_v = 0.05 \cdot (z + z_0)^{5/3} \cdot \Phi_{conv}^{1/3} \quad (24)$$

In the VDI kitchen standard (VDI 1999) which utilizes thermal plume approach, the virtual origin (z_0) is set to be at $1.7 D_h$ below the surface of the appliance. D_h is the hydraulic diameter of the heat source (Eq.3).

When a heat source is located close to a wall, the thermal plume will be attached to the wall. In this case, the entrainment will be reduced compared to the entrainment in a free plume. The air flow rate can be calculated as half of the flow from a source with a heat emission of $2 \times \Phi_{conv}$ (Nielsen 1993).

$$q_v = 0.05 \cdot (z + z_0)^{5/3} \cdot (2 \cdot \Phi_{conv})^{1/3} / 2 = 3.2 \cdot \Phi_{conv}^{1/3} \cdot (z + z_0)^{5/3} \quad (25)$$

If the heat source is located in the corner, the air flow rate is equal to 25 % of the air flow from a heat source with the heat emission of $4 \times \Phi_{conv}$.

$$q_v = 0.02 \cdot (z + z_0)^{5/3} \cdot \Phi_{conv}^{1/3} \quad (26)$$

When several heat sources are positioning close to each other, the plumes merge into a single plume. The total flow from N identical sources is then given by (Nielsen 1993).

$$q_{v,N} = N^{5/3} \cdot q_v \quad (27)$$

It should be noted that the plume follows the Gaussian distribution only after a certain distance. The flow in a buoyant round jet can be divided in three distinct regions where only the last pure plume region follows the Gaussian distribution. In Kofoed's study (1991), the third region starts from the height of 1.8 m above the heat source. Determination of boundaries between the three regions of a buoyant jet is not easy because there is no abrupt change from one region to the other. Therefore, the plume models handle these regions as one group and the model of the plume above the heat source is described by flow of the pure plume region.

Since zero stratification very seldom occurs in real ventilated spaces, the question about the influence of the vertical temperature gradient arises. The effect of the temperature gradient has been studied by several researchers (Fitzner (1991), Jin (1993), Kofoed and Nielsen (1975) and Mundt (1996). In these studies, they have reported that the vertical volume flux from a heat source decreases when the gradient increases. Anyhow based on Mundt (1996) study, the effect of the temperature gradient on the convection flow is very small up to the height of 1 m above the heat source. At the level of 2 m, the effect of the temperature gradient was about 8 % when the heat source and the temperature gradient were 80 W and 3 K/m.

Based on Skåret (Skåret 1986) study, Aumon (Aumon et al 2001) and Bouzinaoui (Bouzinaoui et al 2005) have experimentally studied thermal stratification in ventilated confined spaces. Empirical equations have been proposed for the estimation of the vertical temperature profiles.

It should be noted that in all the published studies the used heat sources are relatively small like a person, a desk lamp, a computer or a heated cylinder. In the commercial kitchen applications, the heat gains are much higher (3 – 60 kW) which means that the effect of the temperature gradient is not so significant than e.g. in the office environment. Nonetheless, the generic model for the effect of the temperature gradient on the convection flow is missing.

4. RESULTS

4.1 Thermal plumes of kitchen appliances during idle mode

The thermal plumes of typical kitchen appliances during idle mode are measured in the laboratory. Based on the conducted measurements, the actual plume characteristics of typical kitchen appliances were studied in paper (I). The results show that the generic plume equation gives reasonable accuracy for practical applications when an individually adjusted empirical factor of the virtual origin is applied.

In the kitchen environment, pollutant fumes from the cooking process are released into the ambient air by the convection plumes. The practical problem is to compute the requested extraction air flow rate to maintain good indoor air quality in an energy efficient manner. Undersized air flow rates could lead to indoor air quality problems and an oversized ventilation system increases unnecessary energy consumption and the life-cycle costs of the ventilation system.

In this study, temperature and air velocity measurements were conducted to characterize the thermal plumes of typical kitchen appliances in a laboratory environment. Both Gaussian distribution and numerical integration methods were used for the establishment of the air flow rate. In the numerical integration method, the flow rate is calculated by summing up the measured velocity samples multiplied by their respective areas. There are pros and cons with both of these methods. The traditionally used Gaussian approximation is accurate if the velocity follows the Gaussian curve. However, close to the appliance the Gaussian approximation is not exactly valid. Direct use of the measured velocities gives an easy platform for air flow calculation but the accuracy of the probes should be sufficient. In this study, both the Gaussian approximation and the direct integration of measured velocities gave almost the same air flow rates.

In the most accurate design method, the design of a kitchen ventilation system is based on the flow rate of the thermal plume. In this method, the amount of heat carried in a convective plume over a cooking appliance at a certain height is calculated. The heat load is then assumed to be a point heat source and the velocity and temperature profiles are approximated to be Gaussian distributed.

The measured convection flows were compared with the generic calculation method of VDI (1999) where the virtual origin is set to be at $1.7 D_h$ below the surface of the appliance. In addition, the effect of the product specific virtual origin on the accuracy of the air flow rate was studied.

The measured and estimated air flow rate of a low capacity induction range is presented in Fig. 15. The results of a more energy intensive iron range, in turn, are presented in Fig. 16. Because the generic plume equation derived in the region of complete flow similarity is not exactly valid in the intermediate zone, the generic power $5/3$ of the distance in the plume equation can not fully describe the induced air flow rate even when the virtual origin is adjusted. With the adjusted virtual origin, it is possible to reach a reasonable accuracy for practical applications. However, the principle of the constant virtual origin as utilized in VDI (1999) is not adequate. The appliance specific factor of the virtual origin should be employed.

The empirical factor of each appliance was adjusted to give reasonable correlation with the measurements. The constant virtual origin approach of VDI standard overestimates the high temperature appliances and underestimates the low temperature appliances. For the high heat gain appliances, the coefficient of the virtual origin is 1.2 – 1.5. For the low heat gain appliance, values of 2.0- 2.5 gives reasonable correlation with the measurements. The location of the product specific virtual origin is strongly dependent on the surface temperature of the appliance. Still, some details of appliances also have a significant effect on the location of the virtual origin.

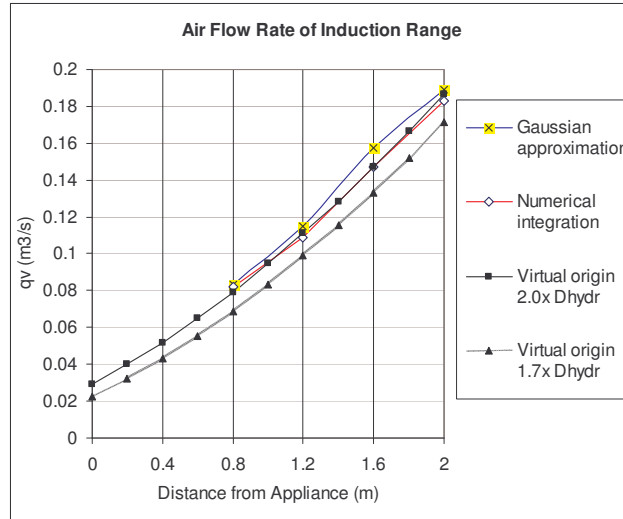


Figure 15. Measured and calculated air flow rates of an induction range.

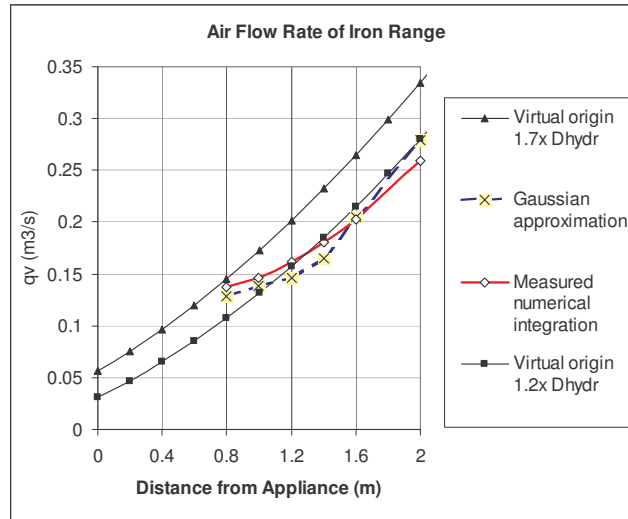


Figure 16. Measured and calculated air flow rates of an induction range.

4.2 Thermal plumes of kitchen appliances during cooking mode

In chapter 4.1, it was shown actual thermal plumes of kitchen appliances. However, all measurements were conducted during idle mode. The objective of the study reported in paper (II) was to analyze the effect of cooking on the plume characteristic and air flow rate. The measurement results give background information to the design productive working

environments where the established air flow rates are sufficient to remove odours and emissions from kitchen appliances.

Based on in tandem study of idle mode conditions (paper I), it can be stated that the same location of virtual origin can be used during idle and cooking modes. The critical factors for accuracy of the equation are the correct estimation of the actual convection load and the optimal selection of the appliance-specific virtual origin. It should be noted that the breakdown of the thermal loads is different during idle and cooking modes. Comparing the results with the idle mode (paper I) reveals that the relative proportion of the convection load is about 5 – 25 %- units smaller.

When boiling water in two kettles, there are two maxima of velocity close to the appliances. These two separate plumes merge at a certain distance, after which the Gaussian approximation gives a good correlation with the measurements. During boiling water in two 10 L kettles, the measured velocity profiles at the heights of 0.8 m and 1.2 m from the an iron range are presented in Fig. 17

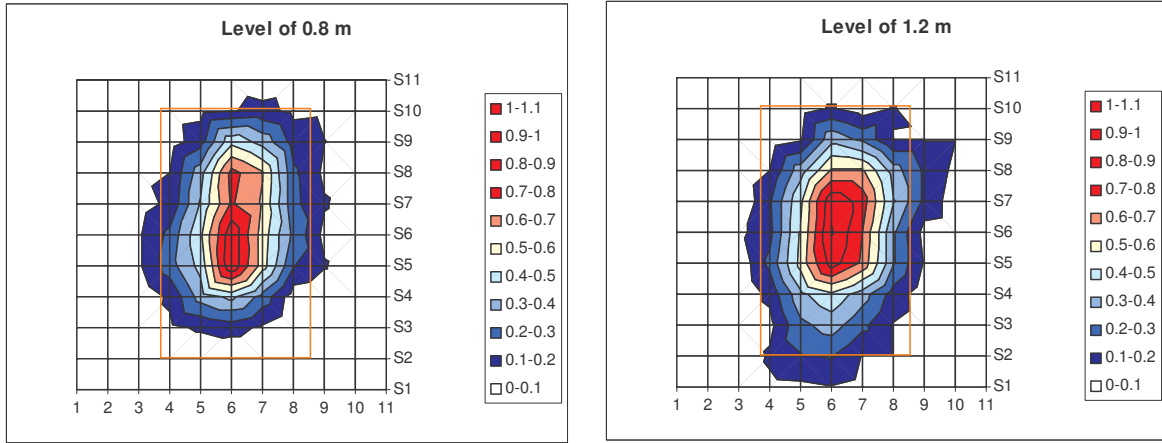


Figure 17. Measured air velocity profiles of an iron range at height of 0.8 m and 1.2 m during cooking mode.

Comparing the parameters p and m during the cooking and idle modes (paper I) indicates a wider convection plume during the cooking conditions. However, the numerical values of the velocity and temperature profiles are more or less of the same order of magnitude. Also, there is no difference in the spreading angle (determined from the growth of R_T and R_v) of the plume between the cooking and idle modes. This means that the cooking process does not have any significant effect on the velocity and temperature distribution of the convection flow.

The reason for that is that the mass flow rate of water during boiling was only about 1 - 1.5 g/s and the volume flow rate of the water vapour is always below 0.9 l/s. Thus, the vapour flow released is small as compared with the induced air flow rate and therefore does not have a significant effect on the convection flow. Thus, the actual convection load and the product specific virtual origin can describe the plume during the cooking process.

4.3 The effect of convection load of the maximum excess temperature, maximum velocity and momentum flux

The basic theory behind the plume calculation has been studied for over 60 years. Basic assumptions about a plume's axial velocity, excess temperature and plume width from a point heat source have been presented e.g. by Schmidt (1941), Rouse et al (1952), Morton (1956), Popiolek (1981) and Kofoed (1991). In previous studies, the used heat sources are relatively small like a person, a desk lamp, a computer or a heated cylinder. In the commercial kitchen applications, the heat gains are much higher (3 – 60 kW) which means that the air velocity and the momentum flux are much higher than in those studies.

In the paper (III), the actual plumes of typical kitchen appliances are presented in both cooking and idle modes. The special emphasis in this study is on the analysis of the correlation between the convection load and the maximum velocity, excess temperature and momentum flux.

From both the idle and cooking mode measurements, the maximum velocity as a function of the distance from the appliance is presented in Fig. 18. According to measurements, the maximum velocity does not significantly decrease as a function of the distance. This shows that the generic plume equation derived in the region of complete flow similarity is not exactly valid. With the relatively high heat gains, the maximum velocity is about 1 m/s at four measured levels (0.8m, 1.2 m, 1.6 m and 2.0 m). With the lower heat gains, the maximum velocity is about 0.4- 0.5 m/s, respectively.

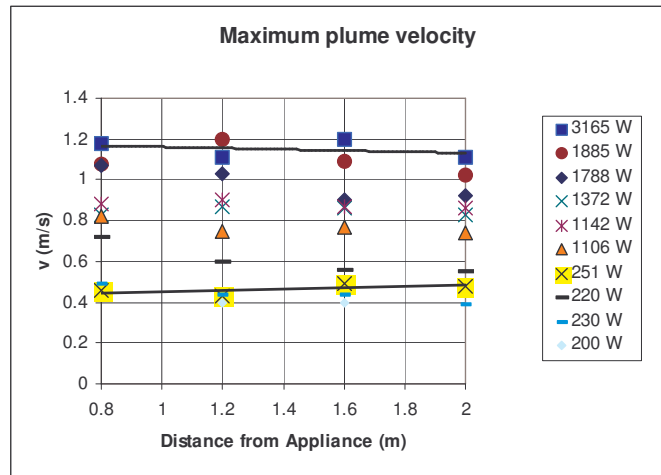


Figure 18. The measured maximum air velocity as a function of the distance from the appliance with different appliances and convection loads.

The maximum temperature difference between the plume and room air as a function of the convection load is shown in Fig. 19. At 0.8 m above the appliance, the temperature difference is about 30 °C with the convection load of 2000 W. Still at the 2.0 m level, the temperature difference remains at about 10 °C.

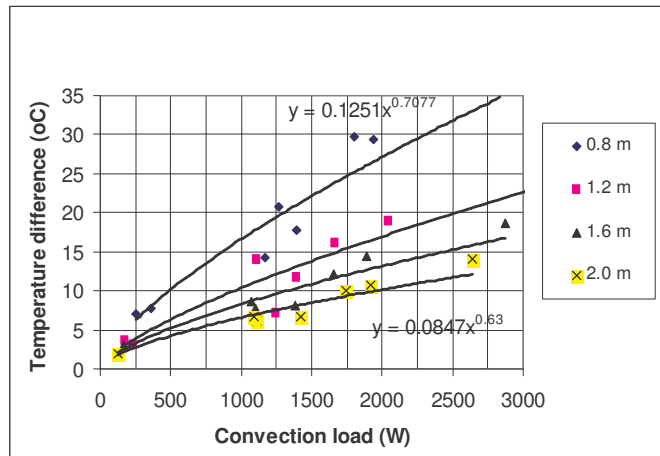


Figure 19. The maximum excess temperature as a function of the convection load at four distances from the appliance

The momentum flux and the kinetic energy flux indicate the strength of the thermal plume. The momentum flux of the thermal plume is a feasible parameter when the interaction between the ventilation system and convection flows is analyzed. In Figs. 20 and 21, the momentum flux and the kinetic energy flux are shown.

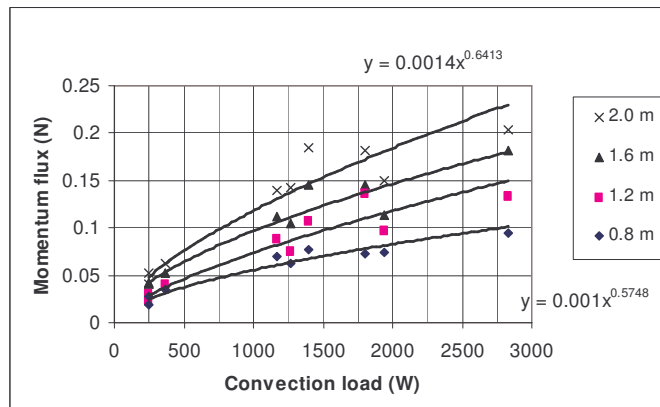


Figure 20. The momentum flux as a function of the convection load at four distances from the appliance

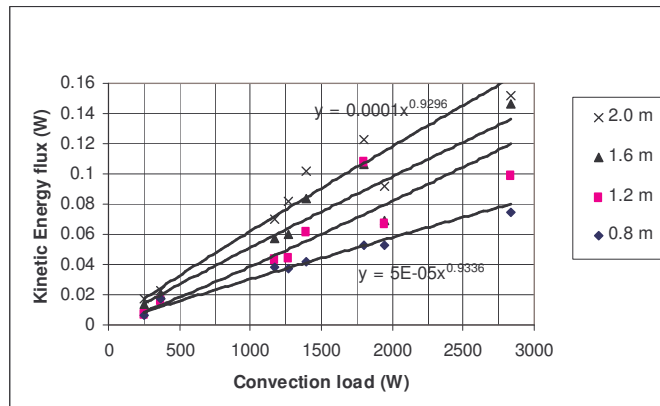


Figure 21. The kinetic energy flux as a function of the convection load at four distances from the appliance

Based on the conducted velocity and temperature measurements, the plume characteristics of the kitchen appliances were computed. Measured values of entrainment, velocity and temperature distribution factors are shown with corresponding values presented in the literature. In Table 5, the following parameters are presented:

- p is the temperature distribution factor
- m is velocity distribution factor
- λ is the ratio factor of temperature and velocity distribution
- α is the entrainment factor

Table 5. Entrainment, velocity and temperature distribution factors of kitchen appliances.

| Author | Plume parameters | | | |
|--------------------------------------|------------------|-------|-------|-----------|
| | α | m | p | λ |
| Popiolek (Popiolek 1981) | 0.036 | 42.5 | 32.8 | 1.14 |
| Schmidt (Schmidt 1941) | 0.124 | 45 | 45 | 1.00 |
| George (George et al 1977) | 0.078 | 55 | 65 | 0.92 |
| Nakagome (Nakagome and Hirata 1976) | 0.103 | 65 | 70 | 0.96 |
| Shepielev (Shiepielev 1978) | 0.045 | 74 | 59 | 1.12 |
| Morton (Morton et al 1956) | 0.093 | 80 | N/A | N/A |
| Rouse (Rouse et al 1952) | 0.036 | 96 | 71 | 1.16 |
| Kofoed (Kofoed 1991) | 0.080 | 110 | 115 | 0.98 |
| Present work idle mode (at 0.8 m) | 0.18-0.26 | 10-23 | 11-25 | 0.95-0.97 |
| Present work cooking mode (at 0.8 m) | 0.11-0.19 | 19-56 | 26-59 | 0.72-0.89 |

In the previous studies of thermal plumes, the velocity and temperature distribution factors are much higher than in the present study. In addition, the entrainment factors of the previous studies are much smaller than in this study. It should be noted that the heat gains from the kitchen appliances are much higher than the measured heat output in the earlier studies. The measurements indicate that the heat gain has a significant effect on the spreading angle and the entrainment factor close to the heat source. The plumes with high heat gains are narrower and the convection flow induces more room air than the previous studies have pointed out.

4.4 The influence of a capture jet on the efficiency of a ventilated ceiling

In the paper (IV), the effect of a capture jet on the efficiency of a ventilated ceiling was evaluated using CFD-simulations. This was supported by laboratory measurements undertaken in another study (Lappeenranta 1994). The laboratory measurements were conducted with and without the capture jet in a simple one-appliance-kitchen layout. The same case-study kitchen is also simulated to obtain a more generic view of the air movement and pollutant levels in the kitchen environment.

The efficiency of the exhaust system is especially emphasised with the ventilated ceiling system where the exhaust is located at ceiling level. The removal efficiency of the total system must be guaranteed and the spread of impurities throughout the kitchen should be prevented.

The efficiency of the exhaust system can be improved with a small capture jet installed at the ceiling surface. The air jet is projected horizontally across the ceiling, which helps to direct heat and air impurities towards the exhaust, Fig. 22. This capture jet represents only about 10 % of the total supply air flow rate. The supply air flow rate is introduced from ceiling units at low velocity.

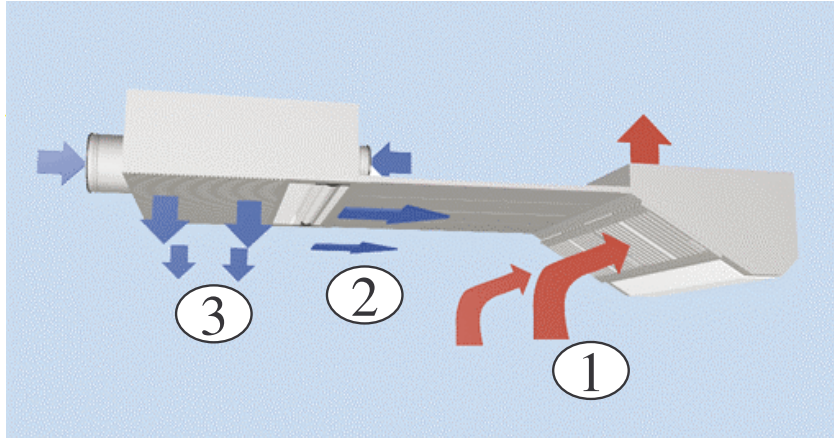


Figure 22. The performance of the capture jet ventilated ceiling. The warm, contaminated air released in the cooking process rises (1) to ceiling level, where it is directed toward exhaust air units equipped with grease filters by the capture air jet (2). The outdoor air flow rate is introduced from ceiling units (3) at low velocity.

Figures 23 and 24 show the velocity fields with and without the capture jet. In the scenario without capture jet (Fig. 23), part of the plume is re-circulated back into the occupied zone. This is due to the ceiling supply, which takes part of the induction air from the plume. This implies a reduction in the efficiency of the extract system. However, in incorporating of the capture jet (Fig. 24) assists the function of the exhaust unit and the plume is extracted effectively without any re-circulation to the kitchen space.

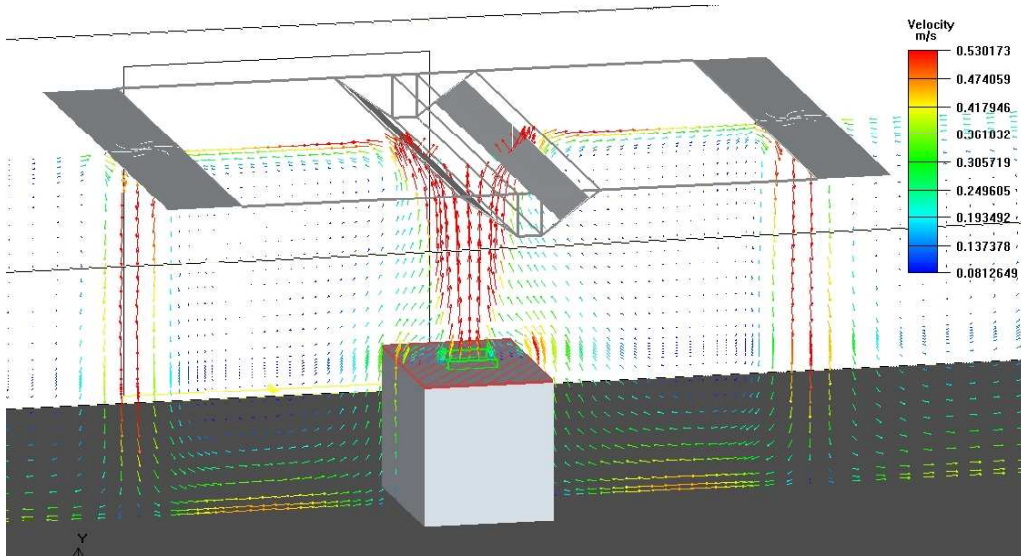


Figure 23. Velocity profile without the capture jet. The velocity vectors show that part of the plume follows the ceiling supply airflow pattern and so returns back to the occupied zone.

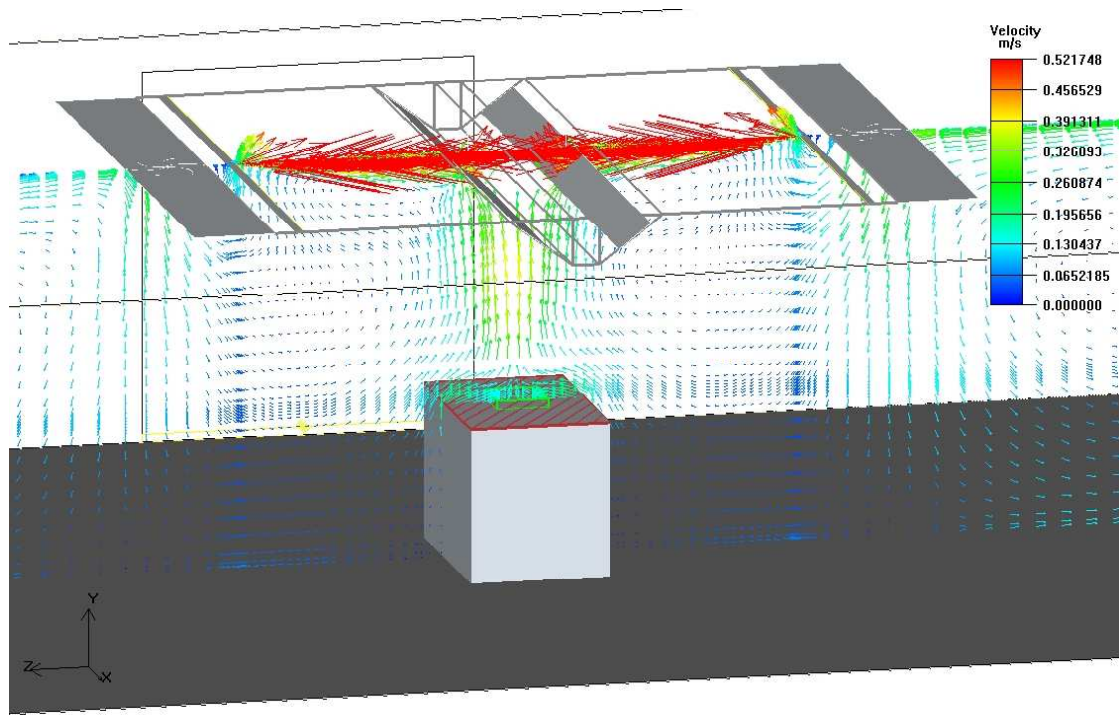


Figure 24. Velocity profile with the capture jet. The velocity vectors show that the capture jet pushes the plume to the exhaust unit. The plume rises up extracted without any re-circulation

Figs. 25 and 26 show the contaminants with and without capture jet. The pollution source in the simulations was water vapor. Fig. 25 shows clearly that part of the plume is re-circulated back to the occupied zone. With the capture jet set-up (Fig. 26), the plume rises directly to the exhaust unit.

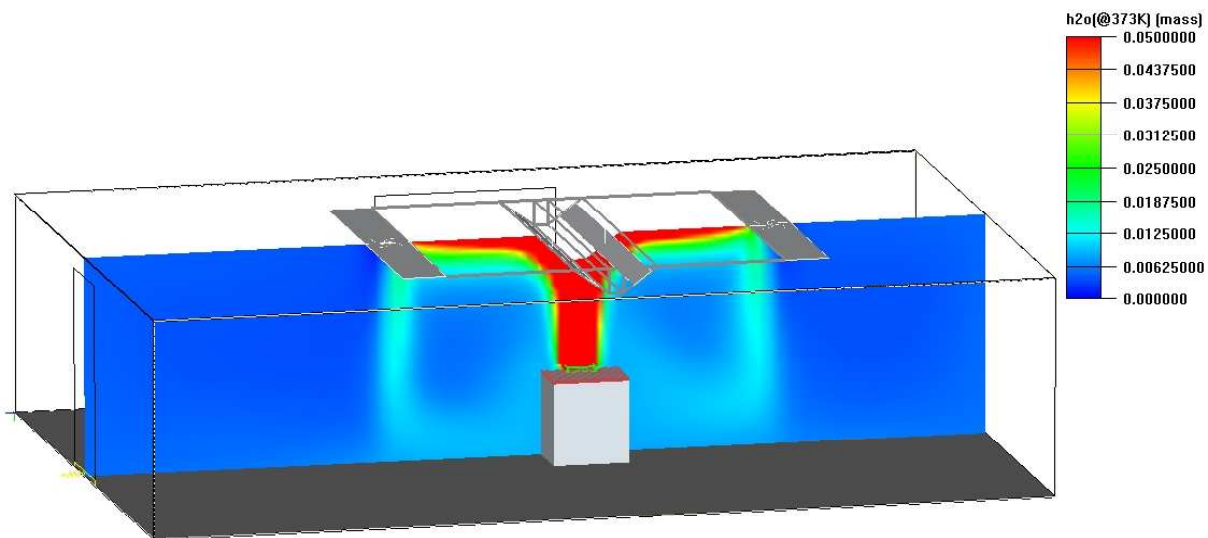


Figure 25. Contaminant level in the kitchen space without the capture jet. A portion of the plume is re-circulated back to the occupied zone

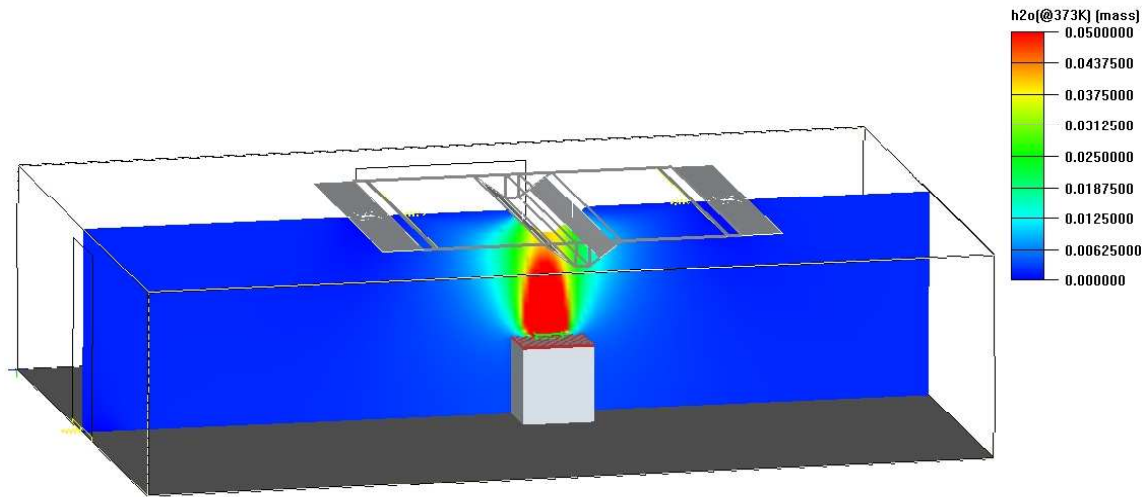


Figure 26. Contaminant level in the kitchen space with the capture jet. The plume is rising nicely toward the exhaust unit.

In a supporting study (Lappeenranta 1994), measurements were conducted with and without the capture jet at three different air flow rates (100 %=design value, 150 % and 50 %). In that study, the concentration of the tracer gas was measured at four locations in the kitchen (Chapter 2.2 Fig. 14). Two sampling points were located underneath the ventilating ceiling in the occupied zone (P1 and P2 at 1.7 m level), and the third point (P3 at 1.7 m) was located outside the ventilating ceiling area. The fourth point (P4) was located near the edge of the ventilating ceiling 0.2 m from the grease extraction unit. Table 6 shows the contaminant distribution in the mock-up kitchen.

Table 6. The measured concentrations with and without capture jet using different air flow rate values

| Design Concept Airflow Rate Capture Jet (on/off) | Concentration (ppm) | | | |
|---|------------------------|-----|----|-----|
| | P1 | P2 | P3 | P4 |
| 50 % (off) | 89 | 113 | 66 | 93 |
| 50 % (on) | 53 | 89 | 42 | 109 |
| 100 % (off) | 21 | 47 | 19 | 20 |
| 100 % (on) | 8 | 10 | 4 | 5 |
| 150 % (off) | 19 | 37 | 12 | 11 |
| 150 % (on) | 7 | 13 | 4 | 4 |

Table 6 shows that, with the capture jet, it is possible to achieve improved indoor air quality in the occupied zone. However, in cases without the capture jet, concentrations in the

occupied zone at point P1 were 2.6 times higher and at P2 were 4.7 times higher, with an air flow rate of 100 %. The concentrations with an air flow rate of 150 % were about 3 times higher. In cases without the capture jet and at 150 % air flow rate, the measured concentrations are higher than in cases with the capture jet at 100 % air flow rate. This shows that even when the exhaust air flow rate is increased by 50 %, the efficiency of the capture jet concept is still better with a much lower air flow rate.

Table 7 presents the simulated contaminant levels at the measurement points (P1-P4) with and without the capture jet. The concentrations are presented with an air flow rate of 100 %. Both the measurements and the simulation give lower contaminant levels when using the capture jet. The only exception is in the simulated case at P1 where the contaminant level is higher with the capture jet. The reason for this difference was a large contamination fluctuation at P1 during the measurement period. Infiltration through the door opening (the space is under pressurised) causes turbulence. It should be noted that this P1 is quite close the range and the boundary of the plume. A minor change in the air movement has a significant effect upon the contaminant level. Simulation results of the capture jet case are shown in Fig. 27 where the measurement points are indicated in three sections that illustrate the contaminant levels around the measured points.

Table 7. The difference between simulated contaminant levels with and without the capture jet concept at 100 % air flow rate.

| CFD Location | Jets on [g/g] | Jets off [g/g] | Difference to jets off case [%] |
|--------------|---------------|----------------|---------------------------------|
| P1 | 0.00937938 | 0.00734761 | 28 |
| P2 | 0.00143387 | 0.012052 | - 88 |
| P3 | 0.00271397 | 0.0044731 | - 39 |
| P4 | 0.00196448 | 0.00563153 | - 65 |

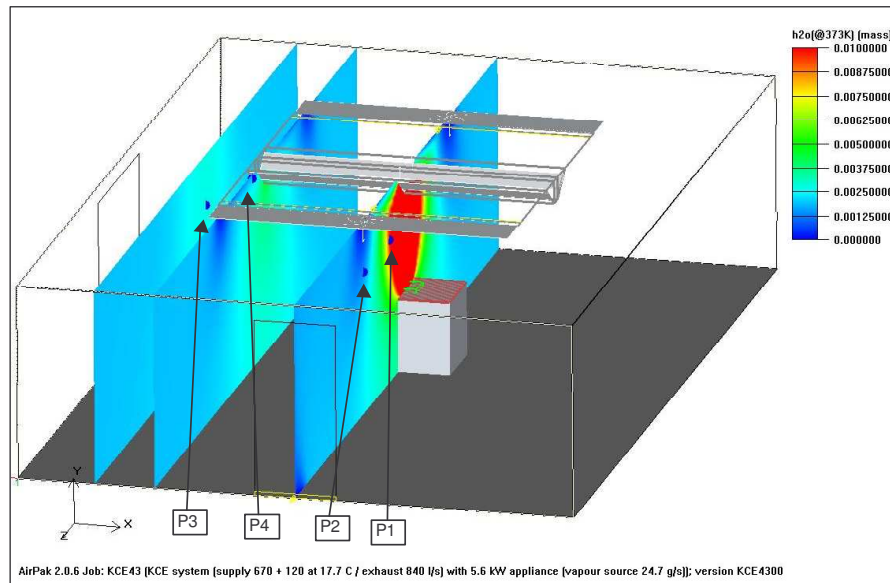


Figure 27. The simulated contaminant levels at the measurements points (P1-P4) with the capture jet in three sections.

Based on the average contaminant level in the occupied zone, it is possible to obtain a more generic view of the total indoor air quality. The average value is a suitable indicator to estimate the level of the pollutant that affect the worker during hours of employment.

The average contaminants are calculated for a 4.8 m x 5.2 m x 1.8 m (H) volume. The central point of the calculated volume is the mid-point of the range. The occupied zone is divided into four different control zones in which average contaminant levels are calculated. In the calculation, the volume just over the range is not taken into account. Figure 28 shows the calculated control zones.

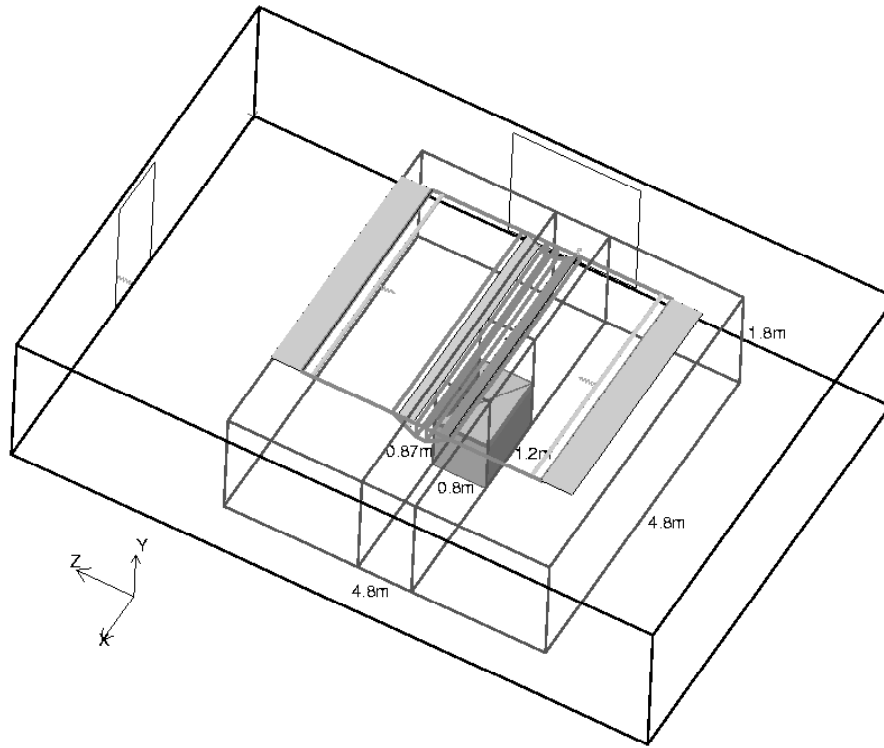


Figure 28. The calculated four control zones in the occupied zone.

Table 8 presents the average contaminant levels in the four control zones and also the average contaminant level in the whole of the occupied zone. In the two control zones parallel with the range (-z and +z), the average contaminant level is about 50 % lower with the capture jet. In the two other zones (-x and +x), the average contaminant level is about 30 % lower.

Table 8. The average contaminant levels in four control zones with and without the capture jet.

| Studied Volume | Jets on [g/g] | Jets off [g/g] | Difference to jets off case [%] |
|----------------|---------------|----------------|---------------------------------|
| - z | 2.823E-03 | 5.597E-03 | - 50 |
| + z | 3.011E-03 | 5.785E-03 | - 48 |
| - x | 3.389E-03 | 4.940E-03 | - 31 |
| + x | 3.933E-03 | 5.439E-03 | - 28 |
| Average | 3.289E-03 | 5.440E-03 | - 40 |

4.5 The capture and containment efficiency of a ventilated ceiling

In this study reported in paper (V), the capture and containment efficiency of a ventilated ceiling is analysed using CFD-simulations. This was supported by laboratory measurements undertaken in another study (Lappeenranta, 1994). The analysis of capture and containment efficiency is a logical continuation of paper (IV) where it is demonstrated that the supply air distribution strategy has a strong influence on pollution removal effectiveness and the thermal environment in kitchens.

Kitchen ventilation standards and design guides do not mention any threshold values for capture efficiency. The main purpose in design practice has been the adjustment of the air flow rate so that it is sufficient to extract the convective heat and contaminants from the occupied zone. The most accurate method is based on monitoring the heat gain of the appliances (VDI 1999). Anyhow, this method is not specially tailored for the kitchen ceiling environment. Also, there is no demands for capture efficiency. In normal design practice, empirical knowledge of existing installations, together with heat load based calculations, have been used for air flow rate determination.

It is common practice to characterise the contaminant removal performance of kitchen hoods in terms of capture efficiency, defined as a ratio between the flow rate of captured contaminant and the total emission rate of contaminants from the source. Although fairly simple in principle, it is not obvious how the capture efficiency of a kitchen extract system should be estimated. There are some practical problems to use that definition, as pointed out in Li and Delsante (1996), in a confined space where there is no general exhaust.

The concept of direct capture efficiency is proposed by Jansson (1990) and Madsen et al. (1994). This approach is also used in the industrial design guidebook (Goodfellow and Tähti, 2001). However, there are measurement and numerical calculation problems in distinguishing the rate of directly captured contaminants from the total captured contaminants, and only an estimation of these factors is possible.

In the engineered type of solution, such as the ventilated ceiling, it not appropriate to separate which part of the contaminant is captured directly. The main idea of the ventilation system is to extract pollution from cooking in order to keep the pollution level in the occupied zone at an acceptable level. It does not matter whether the pollutants are removed directly or indirectly as long as the space condition is within the threshold value.

In this study, with the adaptation of Li's approach, a ventilated ceiling model in a confined space is introduced. In the model, the active height from the floor is set to 1.8 m and the distance from the wall to the appliances is set to 0.3 m (Fig. 29). It should be noted that the overhang of a typical hood is around 25 - 30 cm to cover the dilation angle of the rising plume

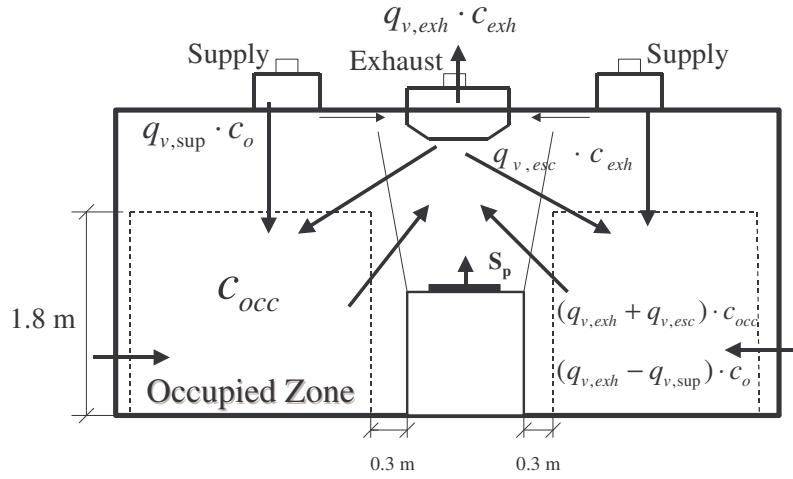


Figure 29. A two zone model for a ventilated ceiling.

In the two zone model, the capture efficiency is then defined as (paper V):

$$\eta_{exh} = 1 - \frac{C_{occ}}{C_{exh}} \quad (28)$$

In the same mock-up kitchen, measurements reported by Lappeenranta (1994) were compared with CFD-simulations. The measured and calculated capture efficiencies for different air flow rates and for two ceiling heights (2.3 and 2.6 m) are presented in Table 9.

Both the measurement and simulated data give lower contaminant levels when the ceiling is at the 2.3 m level. The measured and simulated values of the capture efficiencies compare well for the best capture efficiency range. Outside the optimum range, the difference between the measured and calculated values increases. This could be due to the limited number of measurement points.

The measured and simulated capture efficiencies were 80.8 % and 81.0 % respectively for the 2.6 m ceiling height. However, higher measured and simulated capture efficiencies of 91.3 % and 86.3 % were obtained for the ceiling height of 2.3 m.

It should be noted that increasing the air flow rate will reduce the absolute values of the contaminants, even though the capture efficiency will decrease. The main target should be to maintain the contaminant level at an acceptable level and use the capture efficiency as an indicator of the system efficiency.

Table 9. The measured and calculated capture efficiencies for different air flow rates and two ceiling heights.

| CASES | | MEASUREMENT | | | | | | CFD |
|----------------|------------------|-------------|-------|-------|---------|---------|----------------|----------------|
| Ceiling Height | Exhaust Air Flow | P1 | P2 | P3 | Average | Exhaust | Capture | Capture |
| (m) | q_v (l/s) | (ppm) | (ppm) | (ppm) | (ppm) | (ppm) | Efficiency (%) | Efficiency (%) |
| 2.6 | 500 | 74 | 119 | 74 | 89.0 | 156 | 42.9 | 73.2 |
| 2.6 | 630 | | | | | | (n.a.) | 76.3 |
| 2.6 | 840 | 23 | 16 | 11 | 16.7 | 87 | 80.8 | 81.0 |
| 2.6 | 985 | | | | | | (n.a.) | 74.7 |
| 2.6 | 1090 | 24 | 41 | 11 | 25.3 | 70 | 63.8 | 76.5 |
| 2.3 | 400 | 53 | 89 | 42 | 61.3 | 160 | 61.7 | (n.a.) |
| 2.3 | 840 | 8 | 10 | 4 | 7.3 | 84 | 91.3 | 86.3 |
| 2.3 | 1090 | 7 | 13 | 4 | 8.0 | 64 | 87.5 | (n.a.) |

(n.a.)= Not Available.

4.6 The effect of supply air systems on the efficiency of a ventilated ceiling

The previous studies (papers IV and V) of a one-appliance mock-up have demonstrated that it is possible to improve the capture and containment efficiency with the capture jet. To obtain a more generic view, the laboratory measurements of an appliance block were conducted with different capture jet concepts. As a reference system, the containment removal efficiency of the thermal displacement system was also studied. Based on the conducted measurements, the flush-out factor of the supply air in the theoretical plume equation is derived (paper VI).

The measurements were carried out in a case-study kitchen room with the ventilated ceiling. The layout and the measurement arrangements are presented in Figs. 11 and 12 (Chapter 2.1).

The supply air distribution strategy has a marked influence on pollution removal effectiveness and thermal conditions. The conducted measurements indicate that it is possible to improve the containment removal efficiency of the previously utilized full-length capture jet concept by centralizing the capture jet and exhaust just over the kitchen appliance block. With the same exhaust air flow rate, the efficiency was about 10 % higher than with the full-length concept. Also using the full-length supply, the concentration level close to the appliance block remained at a higher level compared with those points further from the appliances. With the centralized capture jet concept, the concentrations over the working area are almost constant Table 10 presents average concentrations and pollutant removal efficiency with the full-length and centralized exhaust and capture jet concepts.

Table 10. Average concentrations and pollutant removal efficiency with the full-length and centralized exhaust and capture jet concepts. Supply air is introduced from a displacement unit in a reference case. The appliance block is installed close the wall.

| CASES | Concentrations | | Concentrations and Pollutant Removal Efficiency | | | | | | | | Average Efficiency | |
|--------------------------------------|-----------------------|------------------|--|----------------|--------------|----------------|--------------|----------------|--------------|----------------|---------------------------|---------------------------|
| Exhaust Flow Rate | Exhaust P1 | Supply P2 | P3 | Eff. P3 | P4 | Eff. P4 | P5 | Eff. P5 | P6 | Eff. P6 | Average (P3-P6) | Average Efficiency |
| q_{v,exh} (l/s) | (ppm) | (ppm) | (ppm) | (%) | (ppm) | (%) | (ppm) | (%) | (ppm) | (%) | (ppm) | (%) |
| Full-length Case 1/W: 580 | 57.1 | 0.7 | 31.4 | 45.7 | 44.7 | 22.0 | 47.4 | 17.3 | 42.0 | 26.7 | 41.4 | 27.9 |
| Full-length Case 2/W: 700 | 57.3 | 0.91 | 18.0 | 69.7 | 25.8 | 56.0 | 18.6 | 68.6 | 15.4 | 74.3 | 19.4 | 67.1 |
| Full-length Case 3/W: 850 | 47.9 | 0.5 | 10.6 | 78.7 | 10.7 | 78.4 | 5.4 | 89.7 | 4.7 | 91.1 | 7.8 | 84.5 |
| Centralized Case 4/W: 700 | 73.5 | 2.8 | 11.4 | 87.8 | 11.9 | 87.0 | 11.7 | 87.3 | 10.4 | 89.2 | 11.4 | 87.8 |
| Centralized Case 6/W: 850 | 78.9 | 1.6 | 5.9 | 94.5 | 7.6 | 92.2 | 7.0 | 93.0 | 5.9 | 94.5 | 6.6 | 93.6 |
| Displacement Case 8/W: 850 | 79.2 | 1.6 | 2.5 | 98.9 | 2.5 | 98.8 | 2.6 | 98.8 | 3.2 | 98.0 | 2.7 | 98.7 |

Thermal displacement ventilation has the best containment removal efficiency. It was possible to obtain a high ventilation efficiency (98 %) .The average efficiency was about 5 % higher than with the centralized capture jet concept. However in practice, the utilisation of a thermal displacement ventilation system may be difficult because of space constraints in the kitchen. However, if the space permits, the thermal displacement ventilation method must be the first option for the supply air solution. For the design method, it is important to specify the factor of the supply air on the requested exhaust air flow rate that is specially valid for the ventilated ceiling system. That takes into account the interaction between the convection load and the supply air.

The plume equation gives a platform to calculate the air flow rate that is theoretically required to remove the convective heat output of the appliance block. In this study, the flush-out factor of the supply air on the theoretical plume equation was derived for the centralized capture jet concept.

Based on the conducted measurements of the centralized capture jet concept, it is possible to derive a correlation between the air flow rate and the contaminant removal efficiency by employing the theoretical plume equation. The ceiling height of 2.7 m in this study leads to 491 l/s in the wall and 780 l/s in the island type of installation.

Fig. 30 presents the ratio of the used airflow rate and theoretical convection flow (flush-out factor) as a function of the containment removal efficiency. To obtain 85 % and 90 % containment removal efficiency leads to a flush-out factor of 1.2 and 1.5. For practical design work, the target for the containment removal efficiency should be 85 %. For typical kitchen ventilation requirements this provides good indoor air quality with no unnecessary increase in system air flow rates. This selected target for the efficiency means that 15 % of the convection load is released into the room space. This should be taken into account in the cooling load calculation.

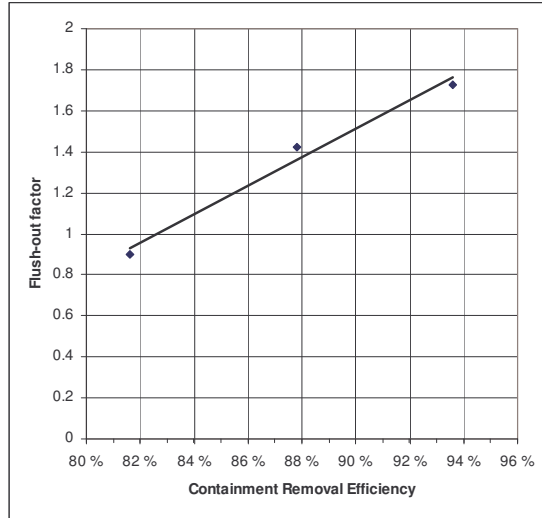


Figure 30. The flush-out factor of the supply air as a function of the containment removal efficiency.

5. DISCUSSION

The starting point for this research was to study the effect of the thermal plumes and supply air systems on the efficiency of a ventilated ceiling. Special attention was paid to analyzing the effect of a capture jet on the contaminant removal efficiency. In that capture air concept, the air jet is projected horizontally across the ceiling, which helps to direct heat and air impurities towards the exhaust. From the practical point of view, the objective of this study was to develop a design process to compute the required air flow rate more accurately.

Laboratory measurements of thermal plumes, tracer gas measurements of ventilation efficiency and computational fluid dynamics simulations of the mock-up kitchen were conducted in order to study the actual plumes of typical kitchen appliances, the capture efficiency of a ventilated ceiling and the effect of the supply system on the efficiency of a ventilated ceiling.

In the thermal plume measurements, the main objective was to measure the air flow rate of thermal plume. The basic measurement grid of 1.1 m x 1.1 m consisted altogether of 121 measurement points (0.1 m interval) in each plane. In this study, the effect of the averaging time on the plume parameters was not studied. The used averaging time was 60 s. The selected averaging time was a compromise between the accuracy and the requested total time for measurements. The averaging time of 60 s is not exactly accurate for the analysis in a single measurement point and the plume axis wandering affects on the individual measurement. However, long time continuous measurements cover the whole wandering area of the thermal plume. Thus, the selected measurement arrangement represents proper statistically coverage of the average condition and the air flow rate determined is accurate.

In the tracer gas measurements of the ventilation strategy study, there were only four measurements points in the occupied zone. Specially with the low ventilation air flow rates, the minor change of the location of the sampling point could change the result of an individual measurement. However, the use of average values over the occupied zone gives appropriate information for the comparison of different ventilation strategies.

Differencing schemes are, among other things, characterized by their order formal order of accuracy. A Taylor series expansion approximates the exact function by a polynomial. A differencing scheme of a given order is determined from the terms of the Taylor series up to that order. The truncation error is determined by the remaining terms. Thus, for the second-order scheme, the truncation error is smaller than with the first-order scheme. In CFD-simulations, the first-order upwind scheme discretization of governing equations was used, because in this application it gives much better convergence than the second-order scheme.

However, the main idea of the simulations were only to compare different ventilation systems. Thus, the exact numerical solution was not so important in this comparison. The validation of the convection flow was not conducted because the requested information about the velocity and temperature profiles was not available. Regardless, the comparison is carried out with the measured sampling point of the tracer gas in the occupied zone. That gives reasonable congruity between the measurements and simulations. Thus, the analysis of the effect of the capture jet concepts on the containment removal efficiency gives the requested reliability.

In this study, the measured convection flows were compared with the generic plume equation where the virtual origin is constant. In the previous studies cited, the generic power $5/3$ of the distance in the plume equation can not fully describe the induced air flow rate even when the virtual origin is adjusted. In the measured heights, the maximum air velocity is more or less constant although the generic plume theory assumed that the velocity should reduce as a function of the distance from the appliance. This depicts that the flow in all practical applications is still in the intermediate zone where complete flow similarity is not valid.

To enhance the accuracy of the air flow rate calculation, a novel plume equation should be developed that is specially derived for the intermediate zone. That model should also be applicable for the design of a displacement ventilation system where the generic plume theory is exploited.

In all the published studies, the used heat sources are relatively small like person, desk lamp, computer and heated cylinder. In the commercial kitchen applications, the heat gains are much higher (3 – 60 kW). The power intensity of the heat gain has much more significant effect on the plume characteristic than the previous studies indicate. The plumes are more narrow and the spreading angle is smaller with relatively high heat gains.

Nevertheless, it is still possible to reach a reasonable accuracy for practical applications with the adjusted virtual origin. The location of the product specific virtual origin is strongly dependent on the surface temperature of the appliance and some details of appliances have also a significant effect on the location of the virtual origin. This means that the exact location of the virtual origin can only be determined if the product specific information is available either from manufacturers' data base or its location of the virtual origin is determined with field measurements.

In the previous studies, the heat gains are studied in the idle mode without any effect of the cooking process. Based on the conducted measurements, the cooking process does not have any significant effect on the velocity and temperature distribution of the convection flow. The reason for that is that the mass flow rate of water during boiling was still small as compared with the induced air flow rate and therefore does not have a significant effect on the convection flow. For the practical design process, this means that the actual convection load and the product specific virtual origin can also describe the plume during the cooking process. The same location of the virtual origin can be utilized in both cooking and idle mode.

This work attempts to make a contribution to the understanding of the thermal plumes of kitchen appliances during idle and cooking modes and the interaction between supply air systems. Further analysis is needed to understand better the effect of dynamic conditions during the cooking process and the plume characteristics close to heat gain.

With the capture jet concept, it is possible to improve the performance of the ventilation system. Even if the exhaust air flow rate without the capture jet is increased by 50%, it is not possible to reach the same contaminant level as that achieved with the capture jet. The conducted measurements indicate that it is possible to improve the containment removal efficiency of the previously utilized full-length capture jet concept by centralizing the capture jet and exhaust just over the kitchen appliance block. Still, the optimization of the performance requires better knowledge of the momentum flux of the heat gain. The capture

jet should be adjusted to be strong enough to turn the convection flow towards to the extract point.

The plume equation gives a platform to calculate the air flow rate that is theoretically required to remove the convective heat output of the appliance block. In this study, the flush-out factor of the supply air on the theoretical plume equation was derived for the centralized capture jet concept. At the moment, there are no standardised target values for capture and containment efficiency, even for kitchen hoods, in any code of practice. In the future, specific target values for the capture efficiency should be determined.

CFD is a promising design tool for the analysis of the interaction between the thermal plumes and supply air systems. However, greater validation would be required to improve the reliability of the analysis. Specifically, validation of the thermal plume's velocity and temperature profiles as well the momentum flux need more research work.

Also, simplified models of the interaction of the convection flows and supply air systems should be developed. In the future, ventilation systems should be capable of optimizing the required exhaust air flow rate based on the actual thermal plume of the appliance block. In this concept, the momentum flux of the capture jet should be adjusted according to rising convection flow. Thus, energy efficiency and indoor air quality are simultaneously enhanced.

6. CONCLUSION

The objectives of this study have been to investigate the actual thermal plumes of kitchen appliances during idle and cooking modes, capture efficiency of a ventilated ceiling and the effect of supply air systems on the efficiency of a ventilated ceiling.

In this study, the measured convection flows were compared with the generic plume equation where the virtual origin is constant. Because the generic plume equation derived in the region of complete flow similarity is not exactly valid in the intermediate zone, the generic power $5/3$ of the distance in the plume equation can not fully describe the induced air flow rate even when the virtual origin is adjusted. Nevertheless, it is still possible to reach a reasonable accuracy for practical applications with the adjusted virtual origin. The constant virtual origin approach overestimates the high temperature appliances and underestimates the low temperature appliances.

The cooking process does not have any significant effect on the velocity and temperature distribution of the convection flow. The reason for that is that the mass flow rate of water during boiling was small compared with the induced air flow rate and therefore does not have a significant effect on the convection flow. Thus, the actual convection load and the product specific virtual origin can describe the plume during the cooking process.

In the previous studies of thermal plumes, the velocity and temperature distribution factors are much higher than in the present study. In addition, the entrainment factors of the previous studies are much smaller than in this study. It should be noted that the heat gains of the kitchen appliances are much higher than the measured heat output in the earlier studies. The measurements indicate that the heat gain has a significant effect on the spreading angle and the entrainment factor close to the heat source. The plumes with high heat gains are narrower and the convection flow induces more room air than the previous studies have indicated.

The efficiency of the exhaust system can be improved with a small capture jet installed at the ceiling surface. The air jet is projected horizontally across the ceiling, which helps to direct heat and air impurities towards the exhaust. In cases without the capture jet, the measured and calculated concentration points in the occupied zone were 2.6-4.7 times higher than with the design air flow rate (100 %). In cases without the capture jet and at 150 % air flow rate, the measured concentrations are higher than in cases with the capture jet and 100 % air flow rate. This shows that even when the exhaust air flow rate is increased by 50 %, the efficiency of the capture jet concept is still better with a much lower air flow rate.

Both the measured and simulated data gave lower contaminant levels when the capture jet was employed. The conducted measurements indicate that it is possible to improve the containment removal efficiency of the previously utilized full-length capture jet concept by centralizing the capture jet and exhaust just over the kitchen appliance block. With the same exhaust air flow rate, the efficiency was about 10 % higher than with the full-length concept. In this study, the flush-out factor of the supply air on the theoretical plume equation was derived for the centralized capture jet concept. For practical design work, the target for the containment removal efficiency should be 85 %. To obtain 85 % containment removal efficiency leads to a flush-out factor of 1.2.

This work attempts to make a contribution to the understanding of the thermal plumes of kitchen appliances during idle and cooking modes and the interaction between supply air systems. Due to the limitations of the conducted study, further research is requested on following topics:

- At the moment, there are no standardised target values for contaminant removal efficiency of kitchen hoods and ventilated ceilings. In future, specific target values for the containment removal efficiency should be determined.
- The conducted measurements show that the flow at the measured heights (0.8 – 2.0 m from appliances) is still in the intermediate zone where complete flow similarity is not valid. To enhance the accuracy of the air flow rate calculation, a novel plume equation should be developed that is specially derived for the intermediate zone.
- All measurement and CFD- analysis are carried out during the steady-state conditions. In future, dynamic situation of the cooking process should be analyzed and the effect on the capture and containment efficiency should be studied.
- In this study, CFD- model of the thermal plume (iron range) was not validated. CFD is a promising tool for analysis the performance of ventilation. However, greater validation is required to improve the reliability of the analysis. Specially, validation of velocity and temperature profiles as well the momentum flux is needed.
- The momentum flux and kinetic energy fluxes are the key parameters that describe the strength of the thermal plume. More research work is required to understanding better the interaction of thermal plumes and jets.

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- I. Kosonen Risto, Koskela Hannu, Saarinen Pekka; Thermal plumes of kitchen appliances: part 1 idle mode
- II. Kosonen Risto, Koskela Hannu, Saarinen Pekka; Thermal plumes of kitchen appliances: part 2 cooking mode
- III Kosonen Risto, Koskela Hannu, Saarinen Pekka; An analysis of thermal plumes.
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- V. Kosonen Risto, Mustakallio Panu; Analysis of Capture and Containment Efficiency of a Ventilated Ceiling.
- VI Kosonen Risto The effect of supply air systems on the efficiency of a ventilated ceiling.